POWER

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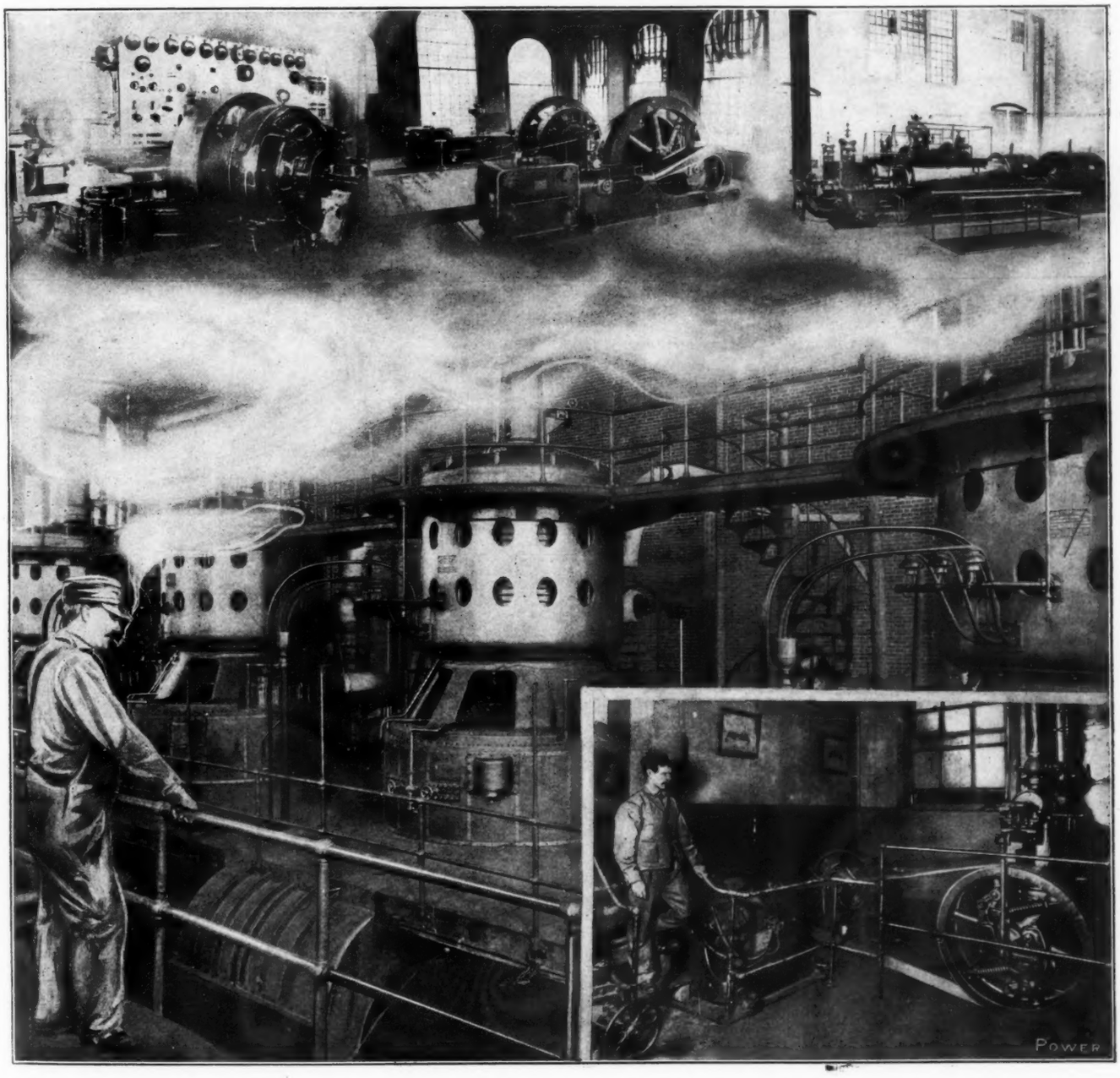
Engineering Progress



THESE TWO ENGINEERS HAD THE SAME OPPORTUNITY: ONE STOOD STILL, THE OTHER PROGRESSED FROM THE SMALL TO THE LARGE POWER PLANT.

Boiler Plant of American Sugar Co.

By Ernest F. Learned

SYNOPSIS—The recently completed boiler plant of the American Sugar Refining Co.'s refinery at South Boston, Mass., was designed with storage facilities for 6500 tons of coal, and an ultimate capacity of 10,160 hp. Some of the features are the design of the structure on a limited site providing for the storage of the coal above the boiler room; unusual natural lighting; the protection from heat of the columns supporting the coal bin in the boiler room, and the use of cement-gun concrete in the upper part of the building.

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The plant, consisting of a boiler room and a coal pocket, adjoins a dock bordering the entire length of the company's property along Fort Point Channel, an arm of Boston Harbor, and is inclosed on three sides by buildings. The structure is 170 ft. long, about 98 ft. wide, and 101½ ft. high. It is of fireproof construction, built of concrete, steel and tile blocking, and is supported on reinforced-concrete foundations carried on piles cut off 6 ft. above mean low water. The boiler room, occupying the main story of the building, is 32 ft. 6 in. in height, its floor being 11 ft. above the level of the grounds.

The coal pocket is located on a longitudinal axis directly above the boiler room, and is 48 ft. wide, 26 ft. high on the sides and 35 ft. 6½ in. at the ends. The lower chords of the roof trusses are carried 23 ft. 6 in. above the top of the bin, allowing a distance of 7 ft. over the peak of the coal when the bin is filled to its capacity. The full width of the building extends for a distance of 103½ ft., the remaining portion at the dock end of the structure being equal in width to the coal pocket.

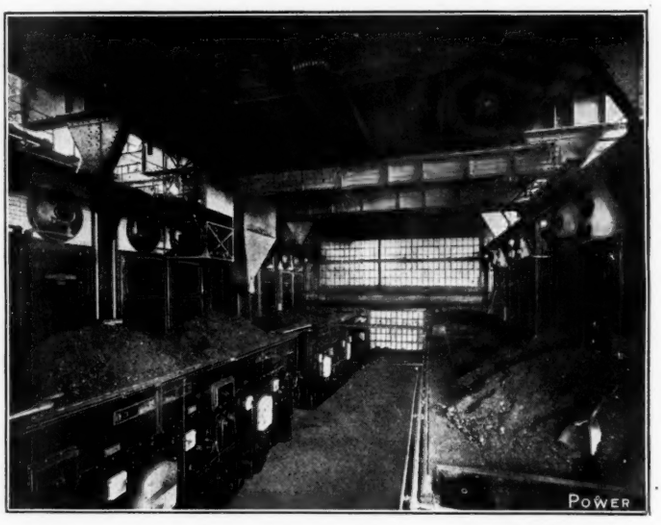

Fig. 1. View of the Boiler Room, Showing Traveling Weigh Hopper

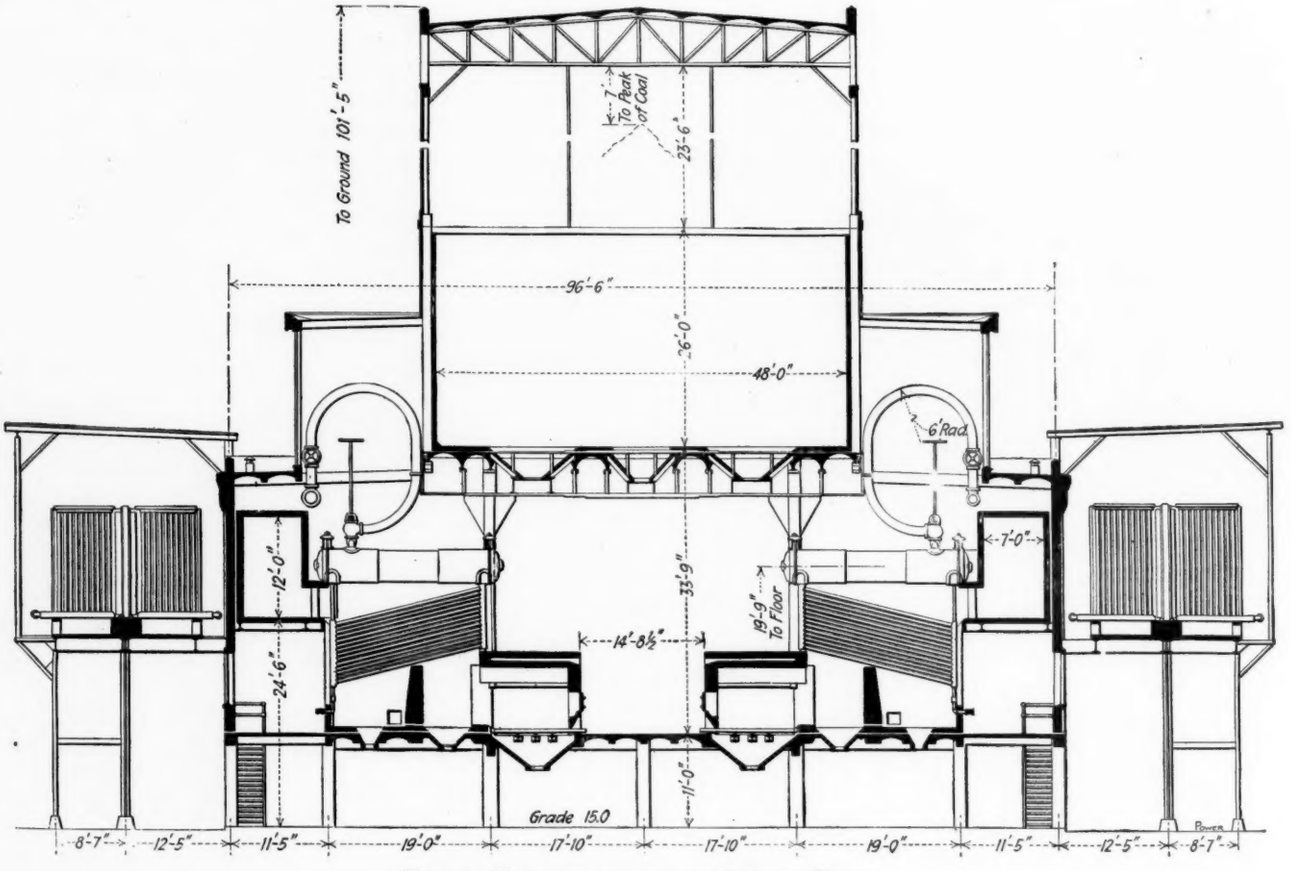

Fig. 2. Elevation of the Boiler Room

The boiler equipment, Fig. 1, consists of twelve 508-hp. Babcock & Wilcox water-tube boilers based on 5080 sq.ft. of heating surface with a ratio of 10 to 1. The boilers are built in batteries of two units each, arranged in two rows facing a single firing aisle 14 ft. 8½ in. wide, the total capacity being 6096 hp. Additional space for four batteries of 1016 hp. each, two on either side, making an ultimate boiler capacity of 10,160 hp., can be obtained by widening the boiler room at its narrow end to its full width. The working boiler pressure is 125 and 80 lb. About 75 per cent. of the steam is used for commercial purposes, the remainder being utilized for power.

The foundations are about 8½ ft. deep and vary in area from 8 ft. square to 17x21 ft., the latter supporting

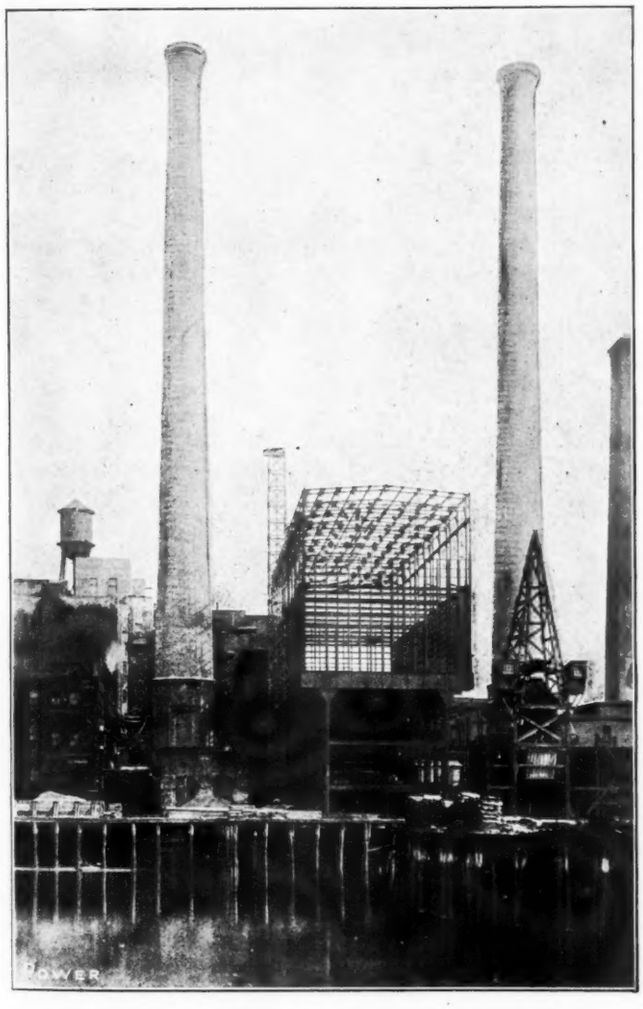

FIG. 3. VIEW OF THE STEEL FRAMEWORK OF THE BOILER HOUSE

two columns. The maximum column load is 467 tons, and is carried on a 13x21-ft. foundation. There are two chimneys, each serving a row of boilers, located midway outside the building adjacent to the side walls. The chimneys are 275 ft. high, 12 ft. in diameter at the top and 15 ft. 8 in. at the base, inside measurements; made of Custodis radial brick; and each rests on a reinforced-concrete foundation 41 ft. square and 10 ft. deep, carried on 400 piles.

The transverse column spacing divides the structure generally into bays 5 ft. 10 in. and 13 ft. 4 in. wide, the smaller being between the boiler batteries, and the larger between the units in a battery.

There are seven longitudinal rows of columns under the main portion of the boiler room, two rows on either side of the center line support the girders carrying the coal pocket. The columns are built of steel encased in

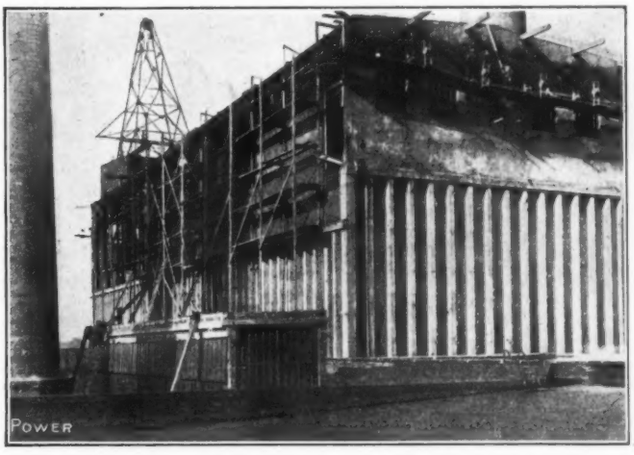

FIG. 4. CONCRETE COVERED COAL POCKET

concrete, except those in the boiler room near the boilers.

The sides of the boiler room not occupied by the windows and columns are inclosed with fireproof tile blocking. The floor is of concrete, arched and supported by I-beams.

In order to provide for natural lighting the coal pocket was built 8 ft. above the boilers, enabling light to pass through the windows on the sides of monitor roofs located on either side of the structure along the lower portion of the coal bin. The windows are 13½x5½ ft. in size, of continuous sash, fitted with wired glass. The expansion loops from the boilers supplying the main header are carried up into the monitor roof, Fig. 2, and the wheel stop valves at either end of the loops can be closed from one position, as the valves attached to the boilers are provided with extension shafts.

A typical girder is 51 ft. 8½ in. long, 5 ft. deep, built of ⅞-in. web plate, 8 and 8 by ⅞-in. angles and 20x½-in. flange plates. These girders act as cantilevers on account of the supporting columns being placed 8 ft. from the ends.

Each boiler equipment includes three reinforced ash hoppers, one being located under the furnace and two under the boiler, as shown in Fig. 2.

On account of the limited space it was necessary to locate the columns within the front line of the boilers in the rear of the Murphy furnaces between the units of a battery, and in order to sufficiently protect the columns from the heat, a 2-in. air space was provided on all sides.

The sides and ends of the bin consist of H-columns and channels placed vertically with T-irons laid horizontally, covered with poultry wire and embedded in a wall of concrete 2 in. thick on each side of the steel.

There are three coal hoppers in each large transverse bay of the bin, making a total of 51 hoppers serving the coal pocket. The coal and ash hoppers are fitted with undercut gates. One row of hoppers is located along the center line of the building, and the other two are about

13 ft. on either side of the center. The steel work between the girders in the narrower bays consists of thirteen 10-in. I-beams spaced generally about 3 ft. 4 in. on centers, and in the larger bays the steel work consists of groups of four longitudinal and four transverse I-beams spaced symmetrically about the hopper locations. The floor, with the exception of the hoppers, is composed of corrugated-iron arches, covered with 7 in. of reinforced concrete. Two 12-in. channels located above each girder tie the sides of the bin at the top. Fig. 3 shows the steel framework of the building.

The roof trusses are carried on H-columns about 28 ft. above the top of the bin, and the sides are inclosed with a 2-in. coating of cement-gun concrete or "gunite" driven against a reinforcement of No. 28 core mesh from the nozzles of a cement gun. The mesh was attached to the columns and to three 5-in. I-beams placed longitudi-

FIG. 5. GENERAL VIEW OF PLANT, SHOWING FUEL-HANDLING APPARATUS

nally between the former. The cement gun working under a pressure of 30 lb. per sq.in. applied the "gunite" composed of one part cement and three parts sand to both sides of the mesh, 1-in. coating on either side. Windows are provided in the upper part of the structure on three sides near the roof, Fig. 4. An advantage in the use of "gunite" is that the column loads are lighter than if the sides were inclosed with tile blocking, such as employed in the lower portion of the building. At the end of the building adjoining the dock there is a 10x14-ft. opening in the upper wall, through which a Robins 30-in. coal conveyor passes from a Procter unloading tower erected on the dock. The roof is reinforced concrete arched between I-beams and covered with tar and gravel.

The unloading tower, Fig. 5, designed for a capacity of 150 tons per hour, is equipped with Meade-Morrison duplex engines and a 1½-ton clamshell bucket. The sizes of the cylinders of the hoisting and trolley engines are 14x24 in. and 10x12 in. respectively. The boom is 124 ft. above the dock and is one of the highest in New England.

The boilers supply steam to the engines through a 5-in. line. The coal is discharged from the bucket into a hopper equipped with a No. 7 Hunt coal crusher reducing its size to 2 in. A Morrick weightometer is installed in the rear of the hopper on a 30-in. conveyor extending the entire length of the coal pocket. The conveyor, suspended from the lower chord of the roof trusses and furnished with a hand tripper, is belt driven at the rate of 400 ft. per min. by a 15-hp., 220-volt, three-phase, 60-cycle motor. The capacity of the conveyor is 150 tons per hour.

Coal is supplied to the furnaces from the bins by means of a 5-ton traveling weigh-hopper running on a track supported by the columns carrying the main girders of the upper part of the structure, as shown in Fig. 1. The weigh-hopper is equipped with a 7.5-hp., 220-volt, alternating-current motor for the longitudinal travel, and with a 2.5-hp. motor for the transverse travel; the respective speeds being 175 and 35 ft. per min. The arrangement of the hoppers and the weigh-hopper permits coal to be taken from any part of the bin and fed to any furnace, thus enabling the removal of the coal regularly from all parts of the bin, and preventing continued storage over an idle boiler and the possibility of an excessive rise in temperature causing spontaneous combustion. The bin can be emptied with the exception of 300 tons without trimming by hand shoveling.

Each row of boilers is provided with a flue and economizer. The flues are of brick 7x12 ft. in section supported on steelwork, extending along the rear of the boilers; and at one end swing outward and run parallel to the sides of the building to the stacks. The economizers are located within the flues near the stacks and consist of two sets of 36 sections, 10 tubes in width. Provision is made for bypassing the flue gases around the economizers. The plant was designed and built by the American Sugar Refining Co., under the direction of F. M. Gibson, chief engineer.

Novel Ash-Handling System

Most coal- and ash-handling machinery is intermittent in its operation, because the storage for coal is generally provided on the boiler-house floor or in bins; ashes are allowed to accumulate in ashpits until a con-

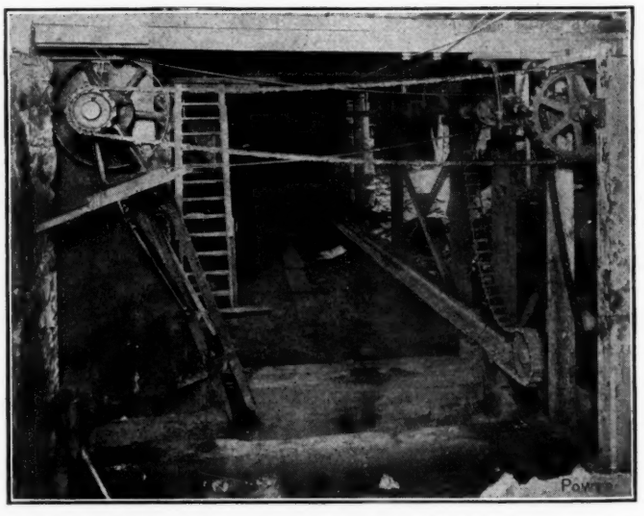

FIG. 1. LINK BELT ASH CONVEYOR

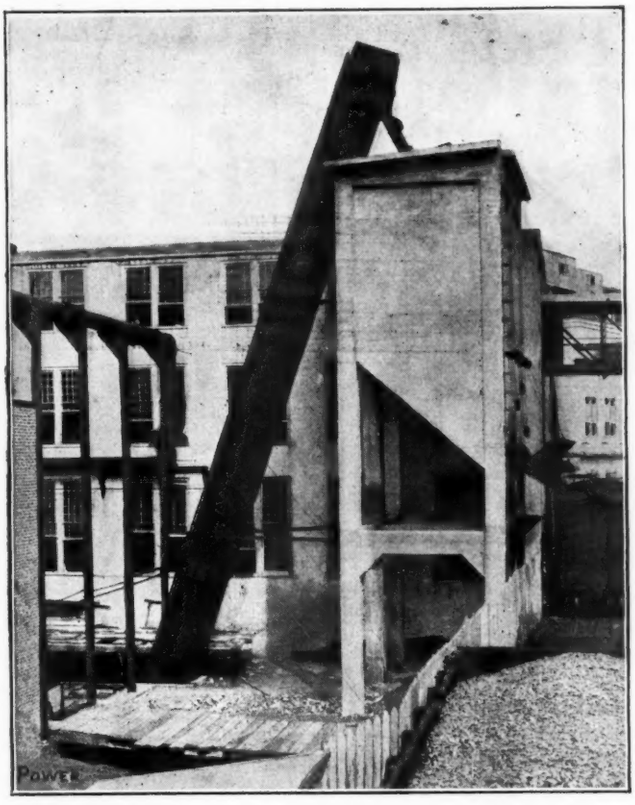

FIG. 2. ASH ELEVATOR DISCHARGING TO CONCRETE ASH BIN

venient time comes to remove them. There are cases, however, where it may be necessary to run the machinery continuously, and to provide against serious wear and rapid deterioration, special designs are necessary.

At the main boiler house of Joseph Bancroft & Sons Co., Wilmington, Del., the ashes which dropped continuously from the stoker grates were allowed to accumulate in a tunnel under the boilers, the tunnel being 11x13 ft. in section, large enough for a horse and cart to go through. In 1911 the boiler house was made longer, and it was decided to install an ash conveyor to remove the ashes from the tunnel as fast as they were dropped, and store them in a bin at one end of the boiler house. This would dispense with the labor of two men and a horse and cart.

In designing the ash-handling machinery, the contractors, the Link-Belt Co., decided to use extremely slow speeds of travel, and to make the parts simple and rugged, so as to show durability while running 24 hours a day.

The ash conveyor, Fig. 1, which runs the length of the ash tunnel, 190 ft., is extremely simple. It consists of a single strand of wide link-belt traveling in a cast-iron trough at 5 ft. per min. The ashes fall from the grates into the lower run, and an empty run slides back overhead. At the end of the run the ashes drop into the boot of an elevator, 84 ft. high, Fig. 2. This elevator travels but 20 ft. per min., and has two strands of Ley Steel-Bushed Chain with heavy malleable-iron buckets. In order to make the buckets discharge at such a slow speed, there are deflector wheels at the head which invert the buckets directly over the head chute. This chute leads to a 4500-cu.ft. reinforced-concrete bin. The steel casing of the elevator carries the weight of all the machinery.

In spite of the difficulties of the service, the ash machinery has shown very little wear, and repairs have been slight. Similar plants have since been erected at other points.

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Standard for Flanges and Flanged Fittings

The use of flanges and flanged fittings has been increasing quite rapidly of late years, in fact, a flange joint is the standard form of connection in many classes of piping. For most purposes the ordinary screw connection should not be used above 6-in. pipe size on account of the difficulty of making and breaking joints.

As the flange-fitting business grew, each manufacturer made up patterns for whatever sizes were called for, to his own personal ideas or knowledge at that time.

These dimensions were later published in catalogs and a wide variation in dimensions was found.

The variation in the diameter and thickness of flanges as well as the bolt circle, size and number of bolts on standard-weight material caused the users and manufacturers considerable confusion and expense.

This brought about the movement by the American Society of Mechanical Engineers and the Master Steam and Hot Water Fitters Association, assisted by the manufacturers of these lines, the result of which was the adoption of what is known as the A. S. M. E. Standard of 1894. This covered flange dimensions and bolting only.

A few years later the manufacturers began to realize that the extra-heavy 250-lb. flanges and bolting were in the same state of confusion. This resulted in the adoption of what has been known as the Manufacturers Standard of 1901.

It became evident in the past few years that the center-to-face and face-to-face dimensions of all flange fittings, including low pressure, standard pressure and extra-heavy pressure, should be established. After considerable work on the part of the M. S. and H. W. F. A. and the A. S. M. E. a standard known as the 1912 U. S. Standard was adopted.

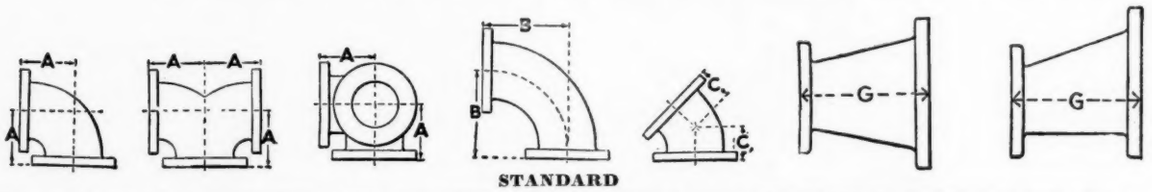
The adoption of this standard opened a discussion among those interested in the subject. It was found that the dimensions presented in the 1912 U. S. Standard differed from the dimensions which had been quite universally used by manufacturers and which covered a large majority of the fittings made.

At a meeting of manufacturers held in New York City, July, 1912, a standard known as the Manufacturer's Standard was adopted by them.

This left the flange-fitting subject in the unfortunate position of there being two standards. When this condition came before the committee of the A. S. M. E., an effort was made by this committee with that of the manufacturer's committee to harmonize, if possible, the differences then existing between these two standards.

After considering the various phases of the subject covering uniformity of design, strength of fitting, strength of bolting as well as the commercial features, a compromise was made.

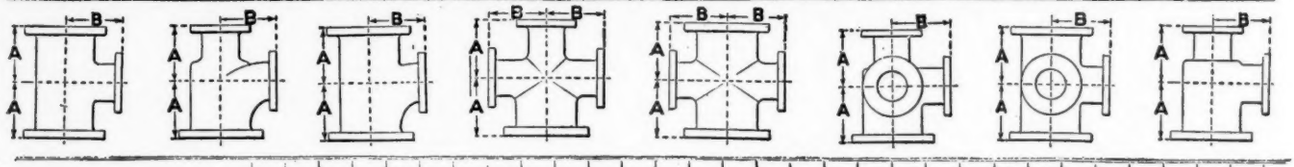
The New American Standard for



STANDARD

Size	1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	9	10	12	14	15	16	18	20	22	24	26	28	30	32	34	36	38	40
AA-Face to Face, Tees and Crosses	7	7½	8	9	10	11	12	13	14	15	16	17	18	20	22	24	28	29	30	33	36	40	44	46	48	50	52	54	56	58	60
A-Center to Face, Ells, Tees and Crosses	3½	3¾	4	4½	5	5½	6	6½	7	7½	8	8½	9	10	11	12	14	14½	15	16½	18	20	22	23	24	25	26	27	28	29	30
B-Center to Face, Long Radius Ells	5	5½	6	6½	7	7¾	8½	9	9½	10¼	11½	12¼	14	15¼	16½	19	21½	22¾	24	26½	29	31½	34	36½	39	41½	44	46½	49	51½	54
C-Center to Face 45° Ells	1¾	2	2¼	2½	3	3	3½	4	4	4½	5	5½	5½	6	6½	7½	7½	8	8	8½	9½	10	11	13	14	15	16	17	18	19	20
D-Face to Face Laterals	7½	8	9	10½	12	13	14½	15	15½	17	18	20½	22	24	25½	30	33	34½	36½	39	43	46	49½	53	56	59					
E-Center to Face Laterals	5¾	6¼	7	8	9½	10	11½	12	12½	13½	14½	16½	17½	19½	20½	24½	27	28½	30	32	35	37½	40½	44	46½	49					
F-Center to Face Laterals	1¾	1¾	2	2½	2½	3	3	3	3	3½	3½	4	4½	4½	5	5½	6	6	6½	7	8	8½	9	9	9½	10					
G-Face to Face Reducers						6	6½	7	7½	8	9	10	11	11½	12	14	16	17	18	19	20	22	24	26	28	30	32	34	36	38	40
Diameter of Flanges	4	4½	5	6	7	7½	8½	9	9¼	10	11	12½	13½	15	16	19	21	22¼	23½	25	27½	29½	32	34¼	36½	38¾	41¼	43¾	46	48¾	50¾
Thickness of Flanges	7/16	1/2	9/16	5/8	11/16	3/4	13/16	13/16	15/16	15/16	1	1 1/16	1 1/8	1 1/8	1 3/16	1 1/4	1 3/8	1 7/16	1 7/16	1 9/16	1 11/16	1 13/16	1 7/8	2	2 1/16	2 1/8	2 1/4	2 5/16	2 3/8	2 7/16	2 1/2
Minimum Metal Thickness of Body	7/16	7/16	7/16	7/16	7/16	7/16	7/16	1/2	1/2	1/2	9/16	5/8	5/8	11/16	3/4	13/16	7/8	7/8	1	1 1/16	1 1/8	1 3/16	1 1/4	1 5/16	1 3/8	1 7/16	1 1/2	1 9/16	1 5/8	1 11/16	1 3/4

Size	42	44	46	48	50	52	54	56	58	60	62	64	66	68	70	72	74	76	78	80	82	84	86	88	90	92	94	96	98	100
AA-Face to Face, Tees and Crosses	62	64	66	68	70	74	78	82	84	88	90	94	96	100	102	106	108	112	116	118	120	124	126	130	134	136	138	142	146	148
A-Center to Face Ells, Tees and Crosses	31	32	33	34	35	37	39	41	42	44	45	47	48	50	51	53	54	56	58	59	60	62	63	65	67	68	69	71	73	74
B-Center to Face, Long Radius Ells	56½	59	61½	64	66½	69	71½	74	76½	79	81½	84	86½	89	91½	94	96½	99	101½	104	106½	109	111½	114	116½	119	121½	124	126½	129
C-Center to Face 45° Ells	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50
G-Face to Face Reducers	42	44	46	48	50	52	54	56	58	60	62	64	66	68	70	72	74	76	78	80	82	84	86	88	90	92	94	96	98	100
Diameter of Flanges	53	55¼	57½	59½	61¾	64	66¼	68¾	71	73	75¾	78	80	82¼	84½	86½	88½	90¾	93	95½	97½	99¾	102	104½	106½	108¾	111	113¼	115½	117¾
Thickness of Flanges	2 5/8	2 5/8	2 11/16	2 3/4	2 3/4	2 7/8	3	3	3 1/8	3 1/8	3 1/4	3 1/4	3 3/8	3 3/8	3 1/2	3 1/2	3 5/8	3 5/8	3 3/4	3 3/4	3 7/8	3 7/8	4	4	4 1/8	4 1/8	4 1/4	4 1/4	4 3/8	4 3/8
Minimum Metal Thickness of Body	1 13/16	1 7/8	1 15/16	2	2 1/16	2 1/8	2 3/16	2 1/4	2 5/16	2 7/16	2 1/2	2 9/16	2 5/8	2 11/16	2 3/4	2 13/16	2 7/8	2 15/16	3	3 1/16	3 1/8	3 3/16	3 1/4	3 5/16	3 3/8	3 1/2	3 9/16	3 5/8	3 11/16	3 3/4



STANDARD

Size	1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	9	10	12	14	15	16	18	20	22	24	26	28	30	32	34	36	38	40
*Size of Outlet and Smaller	All reducing fittings 1″ to 9″ inclusive have the same center to face dimensions as straight size fittings.														6	8	9	9	10	12	14	15	16	18	18	20	20	22	24	24	26
AA-Face to Face, Run															18	20	22	23	24	26	28	28	30	32	32	36	36	38	40	40	44
A-Center to Face, Run															9	10	11	11½	12	13	14	14	15	16	16	18	18	19	20	20	22
B-Center to Face, Outlet															9½	11	13	13½	14	15½	17	18	19	20	21	23	24	25	26	28	29

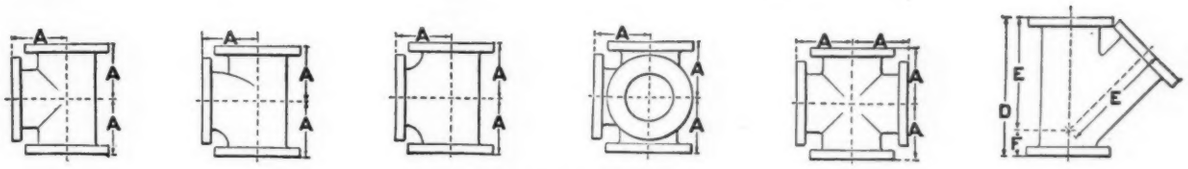
Size	42	44	46	48	50	52	54	56	58	60	62	64	66	68	70	72	74	76	78	80	82	84	86	88	90	92	94	96	98	100
Size of Outlet and Smaller	28	28	30	32	32	34	36	36	38	40	40	42	44	44	46	48	48	50	52	52	54	56	56	58	60	60	62	64	64	66
AA-Face to Face, Run	46	46	48	52	52	54	58	58	62	66	66	68	70	70	74	80	80	84	86	86	88	94	94	96	100	100	104	106	106	110
A-Center to Face, Run	23	23	24	26	26	27	29	29	31	33	33	34	35	35	37	40	40	42	43	43	44	47	47	48	50	50	52	53	53	55
B-Center to Face, Outlet	30	31	33	34	35	36	37	39	40	41	42	44	45	46	47	48	49	50	52	53	54	56	57	58	61	62	63	64	65	67

EXTRA HEAVY

Size	1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	9	10	12	14	15
*Size of Outlet and Smaller	All reducing fittings 1″ to 9″ inclusive have the same center to face dimensions as straight size fittings.														6	8	9	9
AA-Face to Face, Run															18	21	23	23
A-Center to Face, Run															9	10½	11½	11½
B-Center to Face, Outlet															11	12½	14	15

Size	16	18	20	22	24	26	28	30	32	34	36	38	40	42	44	46	48
*Size of Outlet and Smaller	10	12	14	15	16	18	18	20	20	22	24	24	26	28	28	30	32
AA-Face to Face, Run	25	28	31	33	34	38	38	41	41	44	47	47	50	53	53	55	58
A-Center to Face, Run	12½	14	15½	16½	17	19	19	20½	20½	22	23½	23½	25	26½	26½	27½	29
B-Center to Face, Outlet	15½	17	18½	20	21½	23	24	25½	26½	28	29½	30½	31½	33½	34½	35½	37½

Flanges and Flanged Fittings



EXTRA HEAVY

Size	1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	9	10	12	14	15
AA-Face to Face, Tees and Crosses	8	8½	9	10	11	12	13	14	15	16	17	18	20	21	23	26	30	31
A-Center to Face, Ells, Tees and Crosses	4	4¼	4½	5	5½	6	6½	7	7½	8	8½	9	10	10½	11½	13	15	15½
B-Center to Face, Long Radius Ells	5	5½	6	6½	7	7¼	8½	9	9½	10½	11½	12¾	14	15½	16½	19	21½	22¾
C-Center to Face 45° Ells	2	2½	2¾	3	3½	3½	4	4½	4½	5	5½	6	6	6½	7	8	8½	9
D-Face to Face, Laterals	8½	9½	11	11½	13	14	15½	16½	18	18½	21½	23½	25½	27½	29½	33½	37½	39½
E-Center to Face, Laterals	6½	7¼	8½	9	10½	11	12½	13½	14½	15	17½	19	20½	22½	24	27½	31	33
F-Center to Face, Laterals	2	2¼	2½	2½	2½	3	3	3	3½	3½	4	4½	5	5	5½	6	6½	6½
G-Face to Face, Reducers						6	6½	7	7½	8	9	10	11	11½	12	14	16	17
Diameter of Flanges	4½	5	6	6½	7½	8¼	9	10	10½	11	12½	14	15	16½	17½	20½	23	24½
Thickness of Flanges	11/16	3/4	13/16	7/8	1	1 1/16	1 1/16	1 1/4	1 5/16	1 3/8	1 7/16	1½	1⅝	1¾	1⅞	2	2⅛	2 5/16
Minimum Metal Thickness of Body	½	½	½	½	9/16	9/16	9/16	5/8	5/8	11/16	¾	13/16	13/16	⅞	15/16	1	1 1/16	1 1/16

Size	16	18	20	22	24	26	28	30	32	34	36	38	40	42	44	46	48
AA-Face to Face, Tees and Crosses	33	36	39	41	45	48	52	55	58	61	65	68	71	74	78	81	84
A-Center to Face, Ells, Tees and Crosses	16½	18	19½	20½	22½	24	26	27½	29	30½	32½	34	35½	37	39	40½	42
B-Center to Face, Long Radius Ells	24	26½	29	31½	34	36½	39	41½	44	46½	49	51½	54	56½	59	61½	64
C-Center to Face 45° Ells	9½	10	10½	11	12	13	14	15	16	17	18	19	20	21	22	23	24
D-Face to Face, Laterals	42	45½	49	53	57½												
E-Center to Face, Laterals	34½	37½	40½	43½	47½												
F-Center to Face, Laterals	7½	8	8½	9½	10												
G-Face to Face, Reducers	18	19	20	22	24	26	28	30	32	34	36	38	40	42	44	46	48
Diameter of Flanges	25½	28	30½	33	36	38½	40¾	43	45½	47½	50	52¼	54½	57	59¼	61½	65
Thickness of Flanges	2¼	2⅜	2½	2⅝	2¾	2 13/16	2 13/16	3	3⅛	3¼	3⅜	3 7/16	3 5/16	3 11/16	3¾	3¾	4
Minimum Metal Thickness of Body	1¼	1⅜	1½	1 9/16	1⅝	1 11/16	1¾	2	2⅛	2¼	2⅜	2 7/16	2 5/16	2 13/16	2 13/16	2⅞	3

STANDARD

Size	1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	9	10	12	14	15	16	18	20	22	24	26	28	30
*Size of Branch and Smaller	All reducing fittings 1-3½" inclusive have same center to face dimensions as straight size fittings.							2½	2½	3	3	3½	4	4½	5	6	7	7	8	9	10	10	12	12	14	15
C-Face to Face, Run								13	13	14	15	16	16	17	18	20	22	23	24	26	28	29	32	35	37	39
D-Center to Face, Run								11	11	12	13½	14½	14½	15½	17	19	21	22	23	25	27	28½	31½	35	37	39
E-Center to Face, Run								2	2	2	1½	1½	1½	1½	1	1	1	1	1	1	1	½	½	0	0	0
F-Center to Face, Branch								11	11	12	13½	15	15½	16½	18	20½	23	24	25½	27½	29½	31½	34½	38	40	42

EXTRA HEAVY

Size	1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	9	10	12	14	15	16	18	20	22	24
*Size of Branch and Smaller	All reducing fittings 1-3½" inclusive have same center to face dimensions as straight size fittings.							2½	2½	3	3	3½	4	4½	5	6	7	7	8	9	10	10	12
C-Face to Face, Run								14	15	16	17	18	20	21	23	26	29	30	32	34	37	40	44
D-Center to Face, Run								12	12½	13½	14½	15½	17½	18½	20½	23½	26½	27½	29	31	34	37	41
E-Center to Face, Run								2	2½	2½	2½	2½	2½	2½	2½	2½	2½	2½	3	3	3	3	3
F-Center to Face, Branch								13	13½	14½	15½	16½	18½	19½	21½	24½	27½	28½	30½	32½	36	39	43

THE DIMENSIONS GIVEN ABOVE ARE THOSE ADOPTED AS A COMPROMISE BY THE COMMITTEES RESPONSIBLE FOR THE "U. S. 1912 STANDARD" AND THE COMPETING "MANUFACTURERS STANDARD."

FOR EXPLANATION SEE PAGES 527 AND 530

In this compromise the bolting was increased where it was thought necessary. Additional size flanges were added and the center-to-face and face-to-face dimensions were altered to conform to these requirements.

Explanatory Notes

(1) Standard and extra-heavy reducing elbows carry same dimensions center to face as regular elbows of largest straight size.

(2) Standard and extra-heavy tees, crosses and laterals, reducing on run only, carry same dimensions face to face as largest straight size.

(3) If flanged fittings for lower working pressure than 125 lb. are made, they shall conform in all dimensions except thickness of shell, to this standard and shall have the guaranteed working pressure cast on each fitting. Flanges for these fittings must be standard dimensions.

(4) Where long-radius fittings are specified, it has reference only to elbows which are made in two center-to-face dimensions and to be known as elbows and long-radius elbows, the latter being used only when so specified.

(5) All standard-weight fittings must be guaranteed for 125 lb. working pressure and extra-heavy fittings for 250 lb. working pressure, and each fitting must have some mark cast on it indicating the maker and guaranteed working steam pressure.

(6) All extra-heavy fittings and flanges to have a raised surface of $\frac{1}{16}$ in. high inside of bolt holes for gaskets. Standard-weight fittings and flanges to be plain faced. Bolt holes to be $\frac{1}{8}$ in. larger in diameter than bolts. Bolt holes to straddle center line.

(7) Size of all fittings scheduled indicates inside diameter of ports, except for extra-heavy fittings 14 in. and larger when the port diameter is $\frac{3}{4}$ in. smaller than nominal size.

(8) The face-to-face dimension of reducers, either straight or eccentric, for all pressures, shall be the same face to face as given in table of dimensions.

(9) Square-head bolts with hexagonal nuts are recommended. For bolts, $1\frac{5}{8}$ in. diameter and larger, studs with a nut on each end are satisfactory. Hexagonal nuts for pipe sizes 1 to 46 in. on 125-lb. standard, and 1 to 16 in. on 250-lb. standard can be conveniently pulled up with open wrenches of minimum design of heads. Hexagonal nuts for pipe sizes 48 to 100 in. on 125-lb., and 18 to 48 in. on 250-lb. standards, can be conveniently pulled up with box wrenches.

(10) Twin elbows, whether straight or reducing, carry same dimensions center to face and face to face as regular straight size ells and tees. Side outlet elbows and side outlet tees, whether straight or reducing sizes, carry same dimensions center to face and face to face as regular tees having same reductions.

(11) Bullhead tees or tees increasing on outlet, will have same center-to-face and face-to-face dimensions as a straight fitting of the size of the outlet.

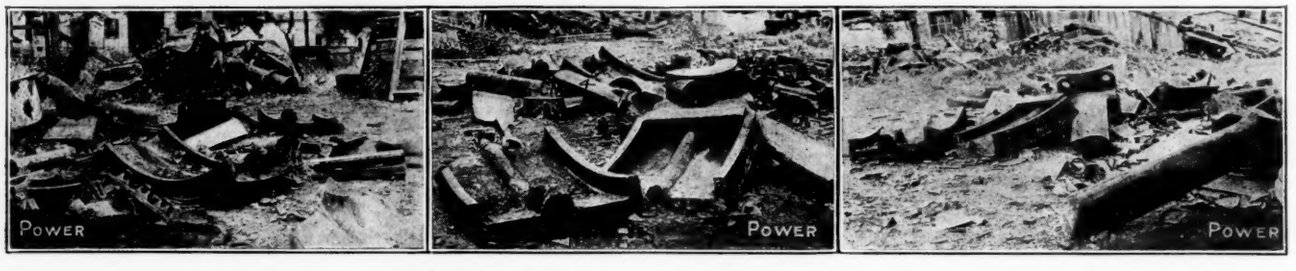
(12) Tees and crosses 9 in. and down, reducing on the outlet, use the same dimensions as straight sizes of the larger port. Size 10 in. and up, reducing on the outlet, are made in two lengths, depending on the size of the outlet as given in the table of dimensions. Laterals $3\frac{1}{2}$ in. and down, reducing on the branch, use the same dimensions as straight sizes of the larger port.

(13) Sizes 4 in. and up, reducing on the branch, are made in two lengths, depending on the size of the branch as given in the table of dimensions. The dimensions of reducing flanged fittings are always regulated by the reductions of the outlet or branch. Fittings reducing on the run only, the long-body pattern will always be used. Y's are special and are made to suit conditions. Double sweep tees are not made reducing on the run.

(14) Steel flanges, fittings and valves are recommended for superheated steam.

Ten-Foot Flywheel Explodes

The accompanying illustrations show the result of a recent flywheel explosion at the power house of the Lawrence Paper Manufacturing Co., Lawrence, Kan. The engine was a 14x36-in. horizontal rolling-mill type, simple noncondensing and ran normally at 120 r.p.m.



The Remains of the 10-Ft. Flywheel

It is right hand, is belted forward and runs under. The 10-ft. flywheel was built in two sections, with a 21-in. face and weighed 9300 lb.

The governor is of the center-weight type and has an automatic stop.

The accident occurred at 10:15 p.m. One piece of the rim, weighing 822 lb., was thrown through the roof and landed 225 ft. distant. Another piece, weight 550 lb., was thrown forward through the engine-room brick wall, up through the roof of the dryer room, across the railway track, coming down in the corrugation department, a distance of 200 ft. This piece fell through a bench where several girls work during the day. Other pieces were found around the engine room. The total weight of all the pieces recovered was 8140 lb.

Some idea of the speed of the engine may be had from the fact that both arms holding the governor weights were bent out at an angle of 45 deg. from the original direction.

It is presumed that the governor was coupled short and could not shorten the cutoff when the load decreased. Fortunately, no one was injured.

Condenser-Tube Corrosion

SYNOPSIS—The lasting quality of condenser tubes is found to depend more upon their microstructure than upon their chemical composition. Slow annealing at a temperature of not over 400 C. after hard drawing produces a close metal of uniform grain which will greatly outlast untreated tubes of the same material.

The committee on steam turbines of the Association of Edison Illuminating Companies, of which committee J. P. Sparrow, chief engineer of the New York Edison Co., is chairman, submitted a report to the annual convention recently held at Cooperstown, N. Y., which contains the most satisfactory and promising treatment of the causes of corrosion in condenser tubes which has come to our knowledge.

The committee classifies tube troubles into three groups:

First—Split or cracked tubes.

Second—Pitted tubes.

Third—More or less complete dezincification resulting in brittle tubes.

The committee assumes that the formerly prevalent idea that corrosion was due to electrolytic action of stray outside currents is no longer given weight, but that local galvanic action set up in the constituents of the alloy by the presence of an electrolyte is the only reasonable explanation of condenser-tube deterioration.

Split tubes are caused by improper mill treatment, either of drawing or annealing. A hard-drawn tube has internal strains, which in many cases results in a split before the tube is placed in service. Cracks at right angles to the axis of the tube may occur when the tube is subject to constant deflection, due to improper support, by vibration, due to steam impact, and consequent hammering on supporting plates, or by constant deflection and rotation, due to friction of steam on the tube surface.

It is practically the unanimous opinion of manufacturers that split tubes may be avoided if the user will accept tubes of the proper temper.

The results of observations made upon tubes in service, and of accelerated corrosion tests, have recently led to the belief that the micro-structure of the alloy had a greater bearing on the corrosion and life of the tube, than the actual composition of the alloy itself. A batch of tubes from one mill, and produced as one lot, would have various lengths of life according to their grain size and hardness.

This fact led to an investigation of the composition, structure and mill treatment of the tubes in question, and it was quite evident that a coarse-grained structure gave short life, that a fine grain, if obtained by hard drawing, gave short life, but that the fine grain of hard drawing, relieved of strain by light annealing, gave evidence of the longest life that could be expected from a particular alloy.

This belief is strengthened by the fact that, with the exception of tube failures from mechanical defects, all failures of coarse-grained tubes come within the second tube-trouble classification of "pitted tubes," while all fine-grained soft-tube failures are, with the same exception as noted, from complete dezincification, tube life in this case being the longest that can be expected under existing conditions of operation.

Investigation showed conclusively that the method of tube production then practiced resulted in very great lack of uniformity, which in all probability accounts for the erratic results noted in service. This is forcibly shown by a plate accompanying the original paper containing ten microphotographs of tube sections taken at random from mill shipments. Five of these were with the ordinary mill treatment, and five where the mill treatment had been such as to secure a uniform product. Our Figs. 1 to 4 are selections from this group of tests, 1 and 2 showing the coarse-grained structure of the ordinary mill process, and 3 and 4 the same material after its structure had been reduced by hard drawing and annealing. The chemical composition of all four tubes is precisely the same (70 Cu 29 Zn 1 Sn), and the life of those which have been treated so that their structure is like that shown in Figs. 3 and 4 is considerably greater than that of the tubes with the coarser structure. The reproductions herewith show a magnification of about 50 diameters.

For salt-water conditions such as prevail about New York, the maximum life of the above mixture (so called Admiralty) is approximately six years. No other copper-zinc alloy has as yet given promise of longer life, although sample tubes of 70-30, 80-20, 90-10 are in service and under observation on the theory that zinc, acting simply as a deoxidizing agent, might best be reduced to minimum quantity on account of its electrochemical activities.

If this theory is correct, it will at once be asked, Why any zinc at all; why not pure copper? The answer is that commercially copper is not pure, but an alloy or solid solution of copper oxide in copper, the oxide being electro-negative.

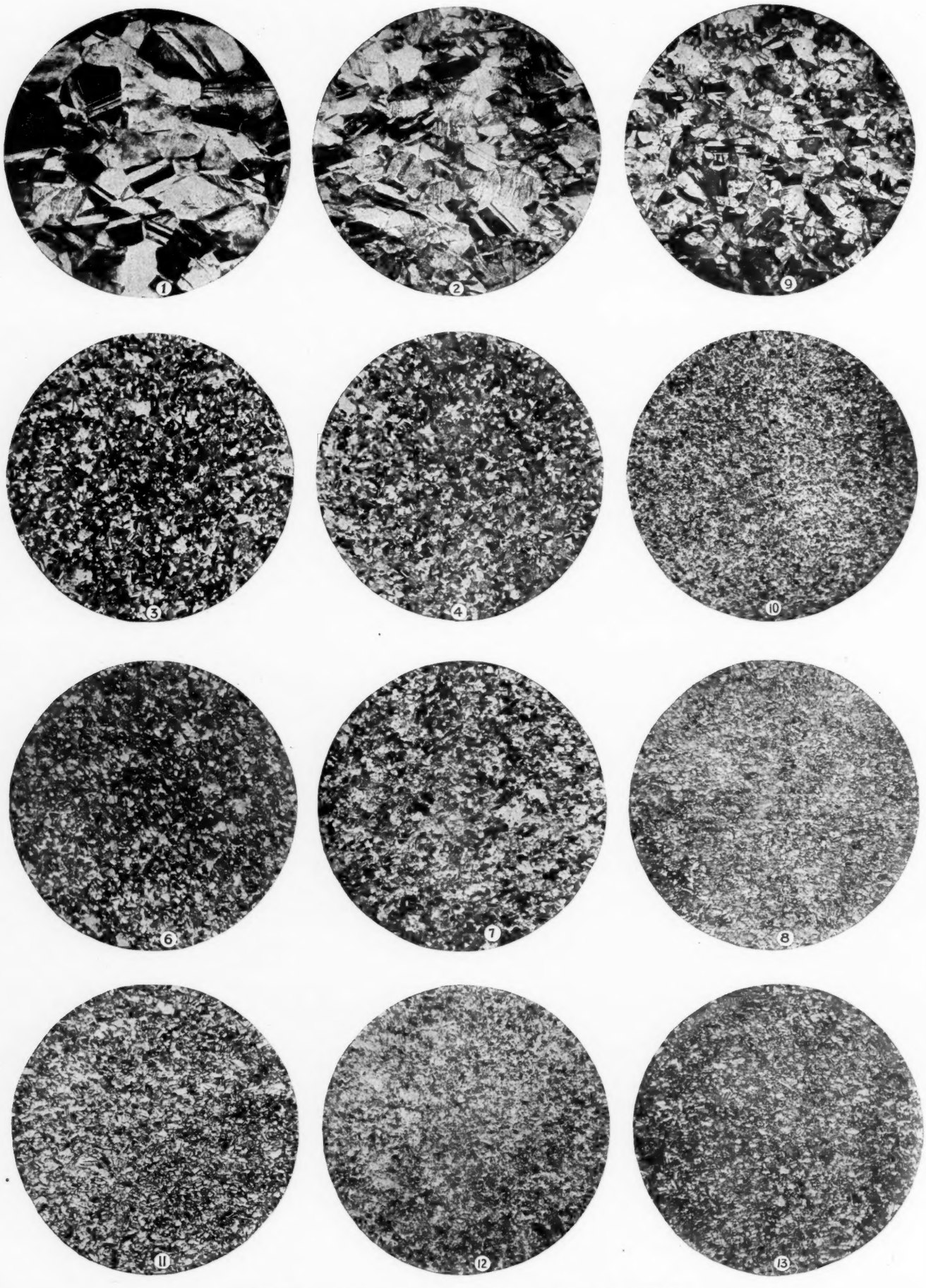
Other alloys than 70-29-1 have been used with varying results; 60-40 or Muntz metal is a standard mixture and has no advantage over the richest copper alloy other than price; 70-30 brass gives similar service to Admiralty and, like this mixture, depends on structure and hardness for long life.

Alloys other than copper-zinc, as Monel metal, 69 Ni 28 Cu 3 misc.; cupro nickel, 28 Ni 72 Cu; aluminum bronze in 8, 5 and 3 per cent. of aluminum have been tried with varying results.

Monel metal is not proof against corrosion in salt water, as is shown by Fig. 5, where the first or left-hand sample shows the result of simple immersion of a Monel metal tube in seawater for a period of 11 months. The central sample shows what happened to the same tube in condenser service for one year and eight months.

The copper-aluminum alloys give promise for salt-water service of being the most durable tube yet produced. Unfortunately they are difficult to manufacture with the higher Al contents. It may be possible that Al acting as a deoxidizer may be fully as effective in the 3 to 5 per cent. mixtures. These are now being tried.

A sample lot of these tubes with an 8 Al, 92 Cu alloy has been in service for 4½ years and shows no signs of deterioration. The right-hand sample in Fig. 5 shows the appearance of one of these tubes after this term of



MICROPHOTOGRAPHS OF CONDENSER-TUBE MATERIALS

service. A second lot of the same mixture failed in four months under the same service conditions, and a third lot now in service gives promise of duplicating the performance of the first.

The reason for the difference in performance evidently lies in the difference in structure as affected by mill treatment. The first lot shows the characteristic structure of the Al bronzes when given a moderate annealing; see Fig. 6. The second lot shows the grain size resulting from heavy annealing, Fig. 7. The third lot shows the effect of light annealing after hard drawing, Fig. 8.

The resulting hardness of the tubes and their ability to resist crushing is practically the same in the first and third lots, Figs. 6 and 8. The second lot, Fig. 7, is softer, with rough longitudinal scratches caused by the scoring of the dies by the oxide of alumina formed by the high temperature of the annealing process.

Figs. 9 and 10 show the structures of 5 and 3 per cent. aluminum-bronze tubes lightly annealed. Structures of 70-30, 80-20 and 90-10 Cu Zn alloys are shown by Figs. 11, 12 and 13 for comparison.

While it is still impossible to draw definite conclusions from the results of the many experiments and investigations under way, it seems reasonable to conclude that

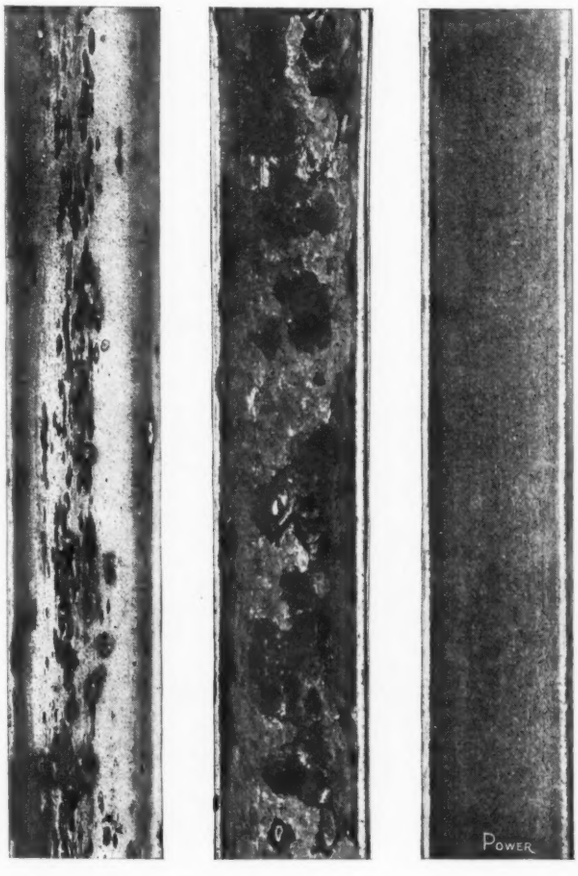


FIG. 5. MONEL METAL AND ALUMINUM ALLOYS

the user may protect himself to the full extent of present knowledge by requiring a manufacturer to furnish:

First—Mechanically perfect tubes, free from slivers and blisters.

Second—Tubes of uniform hardness, as shown from random selection. Test to be deflection between 1-in. square blocks, weighted to secure deflection of 0.005 in. For the 70-29-1 mixture this weighting should be from 400 to 500 lb. for a 1-in. O.D. tube 18 gage.

Third—Uniform grain size in cross-section. This grain size should be fine and secured by final light annealing at low temperature (400 deg. C.) and extended time.

Highly finished tubes are not necessary and indicate

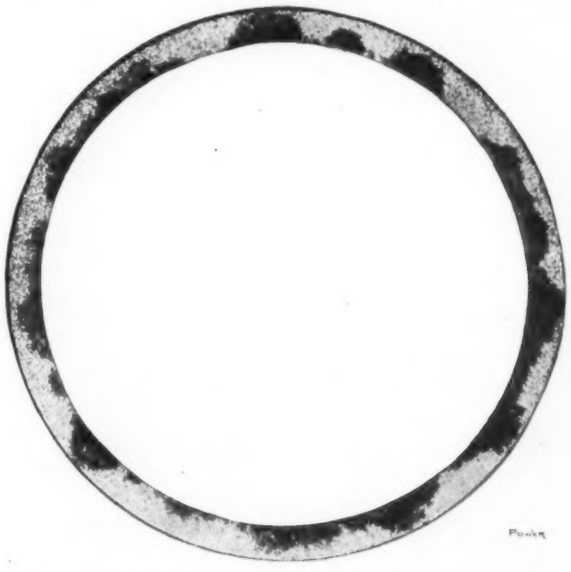


FIG. 14. DEZINCIFIED TUBE 70 CU 29 ZN 1 SN

final hard drawing, producing an ununiform structure.

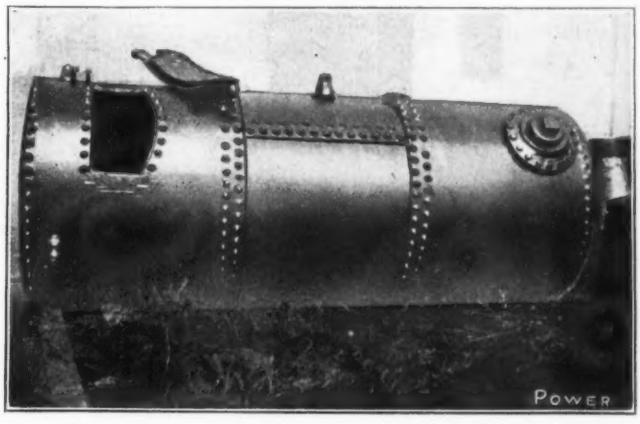
Fig. 14 is a magnified cross-section of a tube having the composition 70 Cu 29 Zn 1 Sn. Dezincification is almost complete; the black areas are spongy copper.

Peculiar Boiler Design

BY GEORGE B. LONGSTREET

While looking over some second-hand boilers recently, I came across the one shown in the illustration.

It is a vertical, fire-tube boiler, 8 ft. 1 in. long by 36



BOILER OF PECULIAR DESIGN

in. in diameter. The shell and firebox sheet are $\frac{3}{8}$ in. thick and the tube sheet $\frac{7}{16}$ in. thick. The stay-bolts are $\frac{7}{8}$ in. in diameter and sustain 42 sq.in. of surface. There are 65 two-inch tubes, 6 ft. long.

The most uncommon features are the double-riveted girth seams and the tube-sheet bracing, thirteen $\frac{7}{8}$-in. stay-rods parallel with the tubes, and headed over.

ELECTRICAL DEPARTMENT

The Induction Motor—I

By F. A. Annett

The almost universal use of electric drive has created a demand for motors to satisfy innumerable conditions. However, to build a motor possessing all the desirable characteristics of the ideal motor is something that the electrical manufacturers recognize as physically impossible, and since the limitations of design make it impossible to construct a universal motor, they are giving more and more attention to the specific requirements of each class of service.

For driving certain kinds of machinery the direct-current motor has no equal, the introduction of the interpole having made it possible to meet still more exacting conditions, where variation in speed and constant torque are required. For this reason almost all direct-driven machine tools and many other devices are driven by direct-current motors either of the shunt or compound types.

There are many localities, however, where only alternating current is available, and unless converted into direct current, which requires more or less complicated and expensive machinery, the use of alternating-current motors is necessary; these are usually of the polyphase or single-phase induction type, polyphase induction motors being used almost exclusively for the larger sizes where polyphase current is available.

For a great many working conditions polyphase induction motors, both of the squirrel-cage and wound-rotor types, possess advantages over direct-current motors; for example, in cement mills, grain elevators, flour mills, saw mills, or any place where it is very dirty and dusty and where a fairly constant speed and good starting torque is required. Also their extreme simplicity, low maintenance charges, solidity of construction and ability to stand hard usage are some of the characteristics that make them an ideal type for use on alternating-current circuits. There are cases of squirrel-cage polyphase induction motors driving pumps in excavations that have been entirely submerged and kept on running until pumped dry without any apparent injury to the motor. Some polyphase squirrel-cage rotor-type induction motors are built impregnated with a special oil and waterproof insulation which results in a winding that can be immersed in water and operated there without damage.

There are, however, localities where it becomes necessary for the central station to supply single-phase current to its patrons. Under such conditions, the single-phase induction motor must be used, also in the smaller sizes it has advantages over the polyphase types; it requires but two wires and one transformer, thus reducing the expense of wiring and transformers about 25 to 30 per cent.

The ever-increasing extension of alternating-current systems has created a demand for motors that will meet the requirements of the many industrial conditions, to meet which the electrical manufacturers have made many improvements in alternating-current motors and are now able to supply them for almost any drive, the operating characteristics comparing favorably with the direct-current motor. The changeable-pole polyphase induction motor and the single-phase commutator-type variable-speed motor both operate through wide ranges of speed depending upon the design, and the compensated single-phase motor operates at approximately unity power factor from half to full load.

If the direction of the current is reversed through a direct-current motor the direction of rotation remains unchanged; hence, theoretically any direct-current motor should operate on an alternating-current circuit. Under this condition not only the armature core, but also the field frame must be laminated to keep the eddy currents down to a minimum. For the motor to operate satisfactorily, however, the current must reverse in the armature and in the field at the same time; that is, the field current must be in phase with the armature current. The simplest way to fulfill this condition is to connect the armature and field circuit in series. The speed, torque and load characteristics of this series commutator type alternating-current motor are distinctly analogous to that of its direct-current prototype operating through wide ranges of speed and torque. On account of having no inherent speed regulation its use is confined either to fixed loads such as fans or blowers, or to varying loads where the motor's controller is constantly under the guidance of an operator. For cranes and railway work this type is admirably adapted. The construction and operation of this type of motor were treated in an article in the Oct. 1, 1913, issue of Power.

A shunt motor will not operate satisfactorily on an alternating-current circuit because the field current lags nearly 90 per cent. behind the armature current. This causes the motor to take a very heavy current from the line, develop a very small torque and spark badly at the brushes. To overcome this difficulty a condenser of such capacity as will bring the field current in step with the armature current may be connected in the field circuit (this has the disadvantage of requiring an almost constant frequency), or the field may be connected to one phase of a two-phase circuit, and the armature to the other phase. This brings the armature current almost in step with the magnetic flux of the field, but this loads the two phases unequally. To overcome this two motors may be used combined in the one structure., the first motor receiving its armature current from the first phase and its field current from the second phase, the second motor receiving its armature current from the second phase and its field current from the first. Such motors have been built and used, but have never met with favor, due to excessive cost in construction, low efficiency, sparking at the brushes, and requiring two-phase circuits. Where polyphase alternating current is available it is in most cases advantageous to use polyphase induction motors.

Polyphase induction motors are usually made to operate on two-phase or three-phase circuits, although they are sometimes operated on single-phase circuits; the last mentioned type will be considered later.

The polyphase motor always consists of two parts, the primary member and the secondary member. The primary member is usually stationary, and consists of a laminated iron ring slotted on its inner surface, in which are placed the primary coils, or as it is usually called, the stator winding. Fig. 1 shows the stationary member of such a motor.

FIG. 1. TYPICAL STATOR

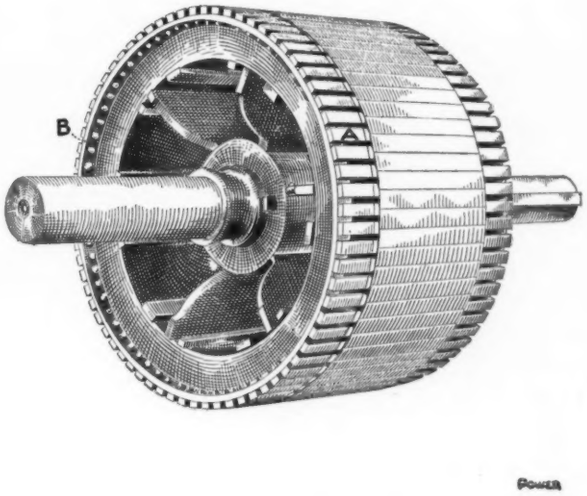


FIG. 2. TYPICAL ROTOR

THE TWO PRINCIPAL PARTS OF A POLYPHASE INDUCTION MOTOR

The secondary member, or rotor, consists of a laminated iron core slotted on the surface. These slots are usually partly closed at the top and carry heavy insulated copper bars, each end connected to a heavy copper ring. Fig. 2 shows the rotor used in the stator shown in Fig. 1. The copper bars are represented at *A* and the short-circuiting ring at *B*.

If the field frame of a direct-current machine should be arranged so that it could be revolved in a set of bearings, the field coils excited, and the commutator on the armature short-circuited with a copper ring, revolving the field frame would revolve the magnetic field about the armature conductors and would have the same effect as revolving the armature conductors across the magnetic field of the pole pieces; that is, the armature conductors will have an electromotive force induced in them, and since they are short-circuited on the commutator, heavy currents will be set up in them. These currents will react upon the field flux and the armature will revolve with the field frame. This is how the rotor of a polyphase induction motor is made to revolve, only the revolving magnetic field is produced electrically and not mechanically as described above; this will be made evident in the following explanation:

FIG. 3a

FIG. 3b

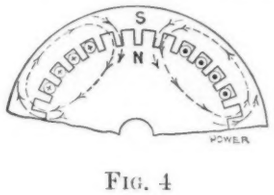


FIG. 4

When an electric current passes through a conductor it sets up a magnetic field which surrounds the wire. The direction of the lines of force bears a definite relation to the direction of the current, that is, when looking along the conductor in the direction the current is flowing the direction of the lines of force will be clockwise. Fig. 3a shows the direction of the lines of force about a conductor when the current is flowing away from the reader and Fig. 3b the direction when the current is toward the reader.

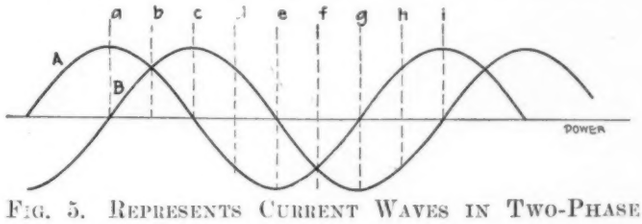


FIG. 5. REPRESENTS CURRENT WAVES IN TWO-PHASE CIRCUIT

The action of the magnetic flux about a group of coils in the stator of an induction motor is shown in Fig. 4. The small circles represent the conductors in section. The conductors marked with a cross represent current flowing away from the reader and those marked with a dot, the current flowing toward the reader. The action of the current in each group of conductors is to produce a magnetic field along the dotted lines as shown by the arrow-heads. From this it is seen that the lines of force

surrounding each conductor coalesce into one field surrounding the entire group and produce north and south poles, as shown.

In Fig. 5, the curves A and B represent the two current waves in a two-phase circuit. That part above the line represents current in one direction and that below the line current in the opposite direction. When curve A is at a maximum, curve B is at zero, and *vice-versa;* that is, the two currents are 90 degrees apart, and if two sets of windings are properly spaced in the stator of an induction motor and connected to a two-phase circuit a revolving field can be produced as the following discussion will show.

Fig. 6 shows the distribution of a six-pole two-phase stator winding, the outside groups of coils belong to one phase and the inside groups belong to the other phase; there are six groups in each phase, or one group for each pole. This winding is similar to the winding shown in Fig. 1, only the coils are placed differently in the slots.

Fig. 7a represents a simple two-phase four-pole winding laid against the inner surface of a ring, A and B being the groups of conductors of each phase, respectively. A and A_1 represent the two sides of one group, A_2 and A_3 represent the two sides of the other group in phase A. B and B_1 represent the two sides of one group and B_2 and B_3 the two sides of the other group in phase B.

If a current is flowing down through the conductors at A it will be flowing up at A_1, similar action taking place at A_2 and A_3, respectively. Referring to the current curves, Fig. 5, and considering the instantaneous values at a it will be seen that the current in phase A is at a maximum, and the current in phase B is at zero. This will give a condition as is shown in Fig. 7a; the current in conductors A will be at a maximum and that in conductors B will be at zero. Assuming that the current is flowing down through the plane of the paper at A and up at A_1, down at A_2, and up at A_3, the conductors with current flowing down are marked with a cross, and those with current flowing up are marked with a dot. Conductors B, B_1, B_2 and B_3 are left blank because there is no current flowing in them at this instant. By referring to Fig. 4, and noting the direction of the lines of force set up about the conductors it will be seen that conductors A, A_1, A_2 and A_3 will have lines of force set up about them in the direction shown by the dotted lines forming four equally spaced poles, north poles at B and B_2, and south poles at B_1 and B_3, as indicated.

At the instant b on the curves A and B, Fig. 5, the currents in the two phases have the same value and direction. In Fig. 7b, assuming the current to be flowing down through the plane of the paper in B and B_2, and up at B_1 and B_3, then the currents in B, B_1, B_2 and B_3 will be in the same direction as the currents in A, A_1, A_2 and A_3, respectively, and the lines of force will encircle the groups A and B, A_1 and B_1, A_2 and B_2, and A_3 and B_3, respectively, in the direction indicated. This

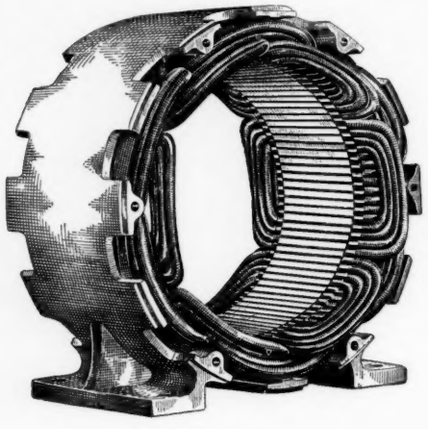


FIG. 6.

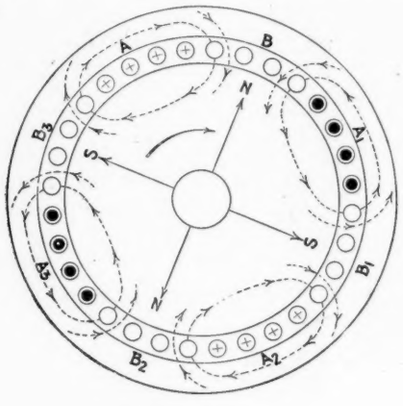


FIG. 7a

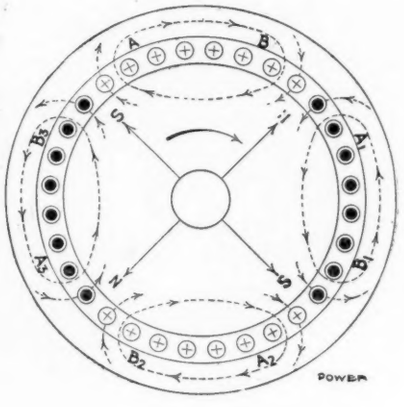


FIG. 7b

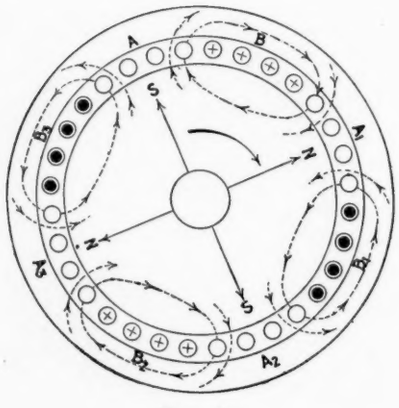


FIG. 7-c

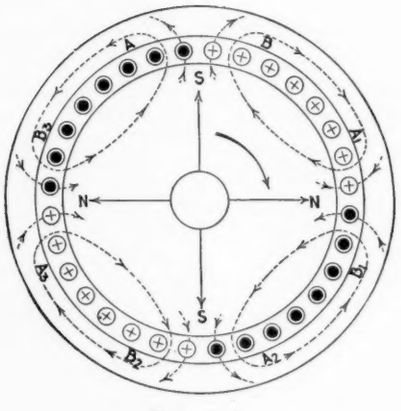


FIG. 7-d

FIG. 7-e

GRAPHIC REPRESENTATION OF ROTATING FIELD

will cause the poles to shift one-sixteenth of a revolution in a clockwise direction as indicated by the arrows.

The current in phase A at the instant c, Fig. 5, is at zero, and the current in phase B is at a maximum value. This will produce a condition as shown in Fig. 7c, and the four poles will have shifted around one-sixteenth of a revolution in the same direction as before.

At the instant d, Fig. 5, the currents in phases A and B are again equal, but that in phase A is opposite in direction. This is just as it should be, as will be seen in Fig. 7d. The current in phase A will flow down through the plane of the paper at A_1 and A_3, and up through the plane of the paper at A and A_2. Lines of force will be set up around the groups of conductors B A_1, B_1 A_2, B_2 A_3, B_3 A_4, respectively, and the four poles will have shifted around one-sixteenth of a revolution in the same direction as before. The position of the poles is as indicated.

The current in phase A at the instant e, Fig. 5, is again at a maximum and that in phase B is at zero. The corresponding positions of the poles are represented in Fig. 7e.

During the period from a to e the current in each phase has passed through one alternation, or one-half cycle, and the poles have shifted 90 degrees, or one-fourth of a revolution, that is, one pole space. In a six-pole machine the poles would have shifted one-sixth of a revolution; in an eight-pole machine one-eighth of a revolution, etc. From this it will be seen that as the currents in the two phases continue to change, the magnetism sweeps around (although the core and coils remain stationary) at a speed per minute which is equal to the alternations per minute divided by the number of poles per phase.

Since one cycle equals two alternations, and the frequency is expressed in cycles per second, the formula for the revolutions per minute may be expressed thus:

$$S = \frac{120\,f}{p} \qquad (1)$$

Where

S = revolutions per minute,
f = cycles per second,
p = number of poles per phase.

As the magnetic field revolves about the rotor, the lines of force are cut by the rotor conductors just as the conductor in the armature of the direct-current machine would cut the lines of force if the field magnets were revolved mechanically, and in the same way the rotor conductors will have an electromotive force induced in them, which will set up currents that react upon the magnetic field of the stator, and will be carried around with it.

Referring again to the direct-current machine, if the armature had been revolved at the same rate as the field magnets the conductors on the armature would not have cut any of the lines of force from the field magnets and would not have an electromotive force induced in them. From this it will be seen that the armature must revolve slower than the field magnets, so that currents may be set up in the armature conductors to develop the necessary torque to carry the load. This is also true of the polyphase induction motor. The speed S referred to in formula (1) is the speed at which the magnetic field revolves, or synchronous speed, as it is usually called. The difference between the synchronous speed and the indicated speed of the rotor is called the slip. In a given motor the slip depends upon the load. At no load the motor runs at very nearly synchronous speed, while at full load in well designed motors the slip should not amount to more than 2.5 per cent. on large motors and 7.5 per cent. on small motors.

If S' represents the indicated speed of the rotor and S the synchronous speed as expressed in formula (1), then the slip equals $S - S'$. Expressing this as a percentage:

$$\textit{Per cent. slip} = \frac{(S - S')\ 100}{S} \qquad (2)$$

For example:

A six-pole motor is connected to a 60-cycle circuit, the indicated speed is 1150 revolutions per minute. Find the percentage of slip.

$$S = \frac{120\,f}{p} = \frac{120 \times 60}{6} = 1200\ r.p.m.$$

$$\textit{Per cent. slip} = \frac{(S - S')\ 100}{S} = \frac{(1200 - 1150) \times 100}{1200}$$

$$= 4.17\ \textit{per cent.}$$

Two Cases of Hunting

The letters by A. N. Kerr and V. K. Stanley on the above subject which appeared in POWER for Sept. 2, are noted. Theirs are evidently cases of the engine governors being oversensitive. In such cases the engines when on load alone will run steadily and keep a constant voltage, but when run in parallel the trouble starts, which usually ends up by one set falling out of step.

The writer has had several cases of this kind and in every instance has been able to overcome it by fitting a fairly powerful oil dashpot to the governors with adjusting screws to regulate the flow of oil from top to bottom of the piston. Such an arrangement is easily fitted, even in the case of the flywheel type of automatic governor, in which case it would be carried off a knuckle joint at its bottom end and fitted with a gland where the connecting-rod leaves the cover. An arrangement similar to this was fitted to some large sets made by the Westinghouse Machine Co. about twelve years ago and put into a London station where they did well.

E. R. PEARCE.

Rochdale, England.

A Giant Merger of Power Systems, four in all, which will furnish power for the operation of industries in western Nevada, the eastern part of California and a large area of southern California, has just been announced by Delos Chappell, president of the Southern Sierras Power Co. The territory to be covered by this system extends from a point as far north and east as Round Mountain in Nevada, to Riverside and surrounding towns.

The various systems will be connected through a line between Bishop and Lundy Lake and the full connection will be effected by the construction of a line from Bishop to Long Valley by the Silver Lake Power & Irrigation Co., and by an extension of the Pacific Power Co.'s lines from Long Valley to Lundy Lake.

The Silver Lake Power Co. has projects on the Owens River, while the Pacific Co., has its plant on Lundy Lake. The parent company is the Nevada-California Power Co., that gets its power from a chain of five hydro-electric power plants on Bishop Creek, which is also the source of the supply of the Southern Sierras Power Co. The latter company also has extensive steam plants at San Bernardino, Calif.

GAS POWER DEPARTMENT

The Farnham Bituminous Producer

Of more than passing interest in the field of suction-gas producers is that developed under the Farnham patents in England and installed in an electric-lighting plant at Skelmorlie, a residential district on the Clyde in Scotland. Although having now been in operation for over two years its construction is perhaps unfamiliar to most American readers.

The producer, as shown in Fig. 1, is of the up-draft type and the working principle eliminates the chance of tarry matter reaching the engines. It consists of a bottom furnace *A*, above which, within the same casting, is a brick-lined generator *B*. At the top of the generator the gas is taken out and is conducted downward on its way to the engine through a pipe surrounded by a water jacket consisting of annular rings, whereby steam is provided to mix with the air necessary for gas production, the air inlet being at the top. The annular jacket *D* surrounds the furnace part of the producer and thus provides a means of rapid steam raising.

A novel feature is the rising and falling grate over which the bituminous fuel is charged and a horizontal sliding plate *E* actuated by gearing, which can be inserted at the bottom of the body of incandescent fuel for supporting it while the furnace is being recharged. Raising and lowering of the grate is effected by the racks *F* and pinions inclosed in such a way as to be out of contact with the ashes or clinker in every position of the grate. Fig. 2 shows the grate in plan and the position of the four elevating-rack rods. The racks and the worms by which the plate is made to travel are either actuated by hand or by an electric motor and speed-reducing gear, the essential parts of which run in an inclosed oil bath.

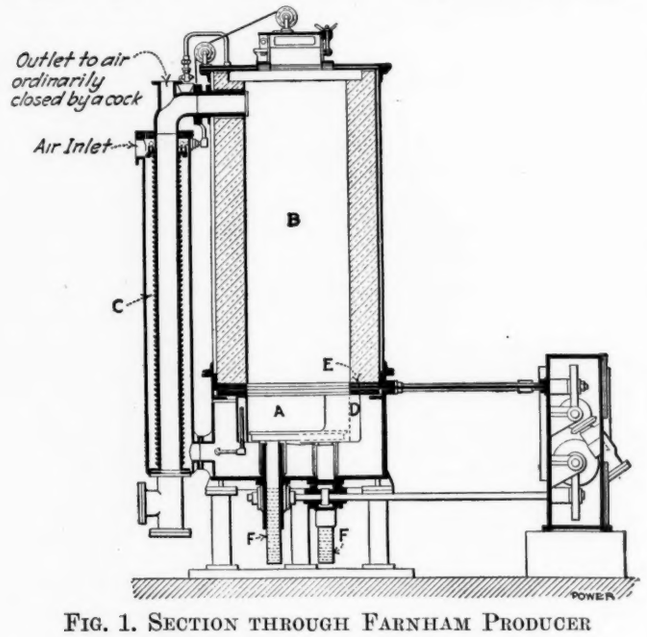


FIG. 1. SECTION THROUGH FARNHAM PRODUCER

To charge the producer, the furnace grate with its weight of incandescent fuel is raised about 10 in. The gear-driven sliding-plate is then introduced between the top of the grate and the bottom of the fuel bed. The whole body of fuel is thus supported while the grate is lowered to its normal position and the fresh charge is inserted. The supporting plate is then withdrawn and the incandescent fuel rests on the fresh supply. The charge, which generally amounts to about 300 lb. in a producer of 100 b.hp., is fed only at intervals of from three to four hours, according to demands upon the producer and the process of charging takes from three to four minutes.

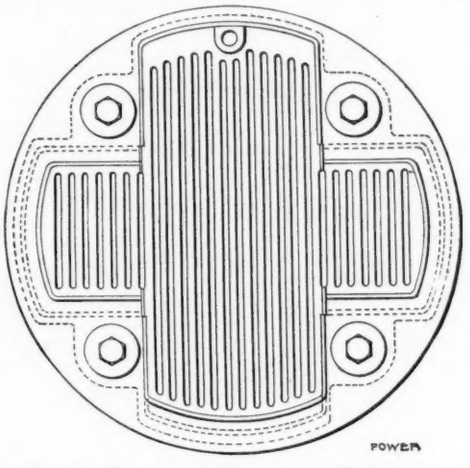


FIG. 2. GRATE OF FARNHAM PRODUCER

As soon as the incandescent fuel falls on the fresh charge the volatile matter is liberated and passing up through the whole depth of superimposed fuel, is decomposed into fixed gases, thus eliminating tar. Moreover, opening the door for charging does not in any way affect the working of the producer or the quality of the gas.

Poking the fuel bed is said to be unnecessary, as the action of raising and lowering the grate compresses the fuel column in such a way that cavities or rifts in the fuel are eliminated. In the opinion of Dugald Clerk, the plant is capable of producing quite clean gas from any bituminous fuel whether of the caking or noncaking variety, as the fuel which he tested was highly bituminous and caking. The consumption of water for all purposes under ordinary conditions of working is comparatively small. For cooling the gas it amounts to 1.17 gal. per hp.-hr., while for the vaporizer only 0.3 gal. per hp.-hr. is required.

The external boiler comes into operation as soon as the gases leaving the producer are hot enough to vaporize the water trickling over and through the perforated ledges surrounding the gas main, the necessary temperature being reached about 1½ hr. after starting. For immediate steam raising after starting, the annular jacket, fitted at the bottom of the producer in contact with the hottest part, comes into operation.

Performance of Motor Ship "Monte Penedo"

After covering over 30,000 miles in regular service between Hamburg, South American ports, and New York, the motor ship "Monte Penedo" was recently inspected at her pier in Brooklyn and the Diesel engines were found to be in excellent condition.

This vessel, it will be remembered, is a cargo boat of 4000 tons gross register and was placed in commission about a year ago. She is equipped with two four-cylinder, two-stroke-cycle Sulzer-Diesel engines each of 850 hp. at 160 r.p.m. and has a speed of about 11 miles per hour. Quoting from the inspector's report:

"In order to inspect a cylinder head and a piston, one cylinder head was taken off and the piston taken out. The wearing surfaces on the piston, piston rings, and especially the cylinder were in perfect condition and the latter was as smooth as a mirror. The parts of the cylinder cover and piston forming the combustion space appeared black, as if covered with a thin coat of paint, but no large carbon deposits were found. Neither was there any appreciable amount in the exhaust ports.

"No troubles were experienced on any of the main bearings, pins or crosshead guides. The parts requiring the most attention were the valves for the cooling water, bilge pumps and air compressor. The fuel-pump packing seems to be tight and no leakage has been apparent."

Pure mineral oil is used for lubricating the cylinders, the consumption averaging about 6⅓ gal. per 24 hr. The lubrication of other parts (including all the auxiliary engines) amounted to about 46 gal. per 24 hr. Sea water is used for the cylinder jackets, piston and oil cooling. The fuel used is a gas oil having a specific gravity of 0.86 to 0.89 and the consumption averaged 15,604 lb., or approximately 2170 gal. per 24 hr.

❋

Horsepower of Small Gas Engines

The method of computing the indicated horsepower of a gas engine running at slow or moderate speed follows so closely the procedure for steam-engine practice that seldom any misunderstanding arises. The principal point to bear in mind is that in the $\frac{PLAN}{33{,}000}$ formula N must represent the number of explosions which in a hit-or-miss governed engine may not bear any definite relation to the revolutions per minute. Moreover, in a gas engine a stiffer spring must be used, owing to the high explosion pressures.

With small high-speed engines running at 800 to 1000 r.p.m. the problem is quite different. Here the speed is too high to obtain accurate results with an ordinary indicator; hence, it is usual to estimate the mean effective pressure. The latter, however, will vary with the load and is dependent upon several factors, including the mixture, the compression pressure, and the time of ignition. Therefore, when running underloaded it is practically impossible to arrive at the horsepower, without actually measuring the brake horsepower. The full-load horsepower may be estimated, with fairly close approximation, however, if the cylinder dimensions and speed are known, by assuming a mean effective pressure for full-load conditions. With tight valves and pistons a mean effective pressure of 80 lb. may be taken for gasoline, 75 lb. for natural or illuminating gas and 65 lb. for producer gas; then substitute this value for P in the formula.

$$Horsepower\ (per\ cylinder)^* = \frac{PLAN}{33{,}000 \times 12}$$

Where

P = Mean effective pressure;
L = Length of stroke in inches;
A = Area of piston in square inches;
N = Explosions per minute.

This may be written:

$$Horsepower\ per\ cylinder = \frac{P\,L\,0.7854\,D^2N}{33{,}000 \times 12}$$

where D = *diameter of cylinder*

Substituting for gasoline:

$$I.hp. = \frac{80\,L\,0.7854\,D^2N}{33{,}000 \times 12} = \frac{D^2LN}{6300}$$

Assuming a mechanical efficiency of 85 per cent.

$$B.hp. = \frac{D^2LN}{6300} \times 0.85 = \frac{D^2LN}{7400}$$

This formula may be expressed in other terms to suit special conditions. For instance, in a four-stroke-cycle engine with jump-spark ignition where an explosion occurs at every second revolution, N may be taken as the revolutions per minute and the formula will become

$$B.hp.\ (per\ cylinder) = \frac{D^2LN}{14{,}800}$$

This is the more familiar form, but it has been further simplified by the American Licensed Automobile Manufacturers Association in assuming a piston speed of 1000 ft. per min., which is equivalent to a 6-in. stroke and 1000 r.p.m. or, 5-in. stroke and 1200 r.p.m. Using the round number 15,000 for the denominator this reduces to:

$$B.hp.\ (per\ cylinder) = \frac{D^2 \times 6 \times 1000}{15{,}000} = \frac{D^2}{2\frac{1}{2}}$$

This is the formula by which automobile engines are rated.

❋

Alcohol as a Decarbonizer

In a paper read at the recent Gas Engine Show, at Kansas City, J. A. Anglada cited some interesting experiments which he had made with denatured alcohol as a carbon-remover in gasoline engines. He found that the best results were obtained when the combustion space of a hot engine is entirely filled with denatured alcohol and allowed to stand for six or eight hours. Although heat is not absolutely necessary, it hastens the loosening of the carbon deposit which will pass out through the exhaust when the engine is started.

In cases where some of the alcohol leaked past the piston into the crank case it rendered the lubricating oil unfit for use, although when run in this way for a very short period the presence of alcohol in the oil appeared to cleanse the bearing surfaces.

After the carbon has been removed in this way a small amount of denatured alcohol (from 2 to 4 oz.) introduced into the combustion space every three or four days, when the engine is in constant use, will keep the cylinders free.

*Strictly speaking, this should be "per cylinder per end," but as the present discussion deals only with small engines, which are always built single-acting, the formula may be understood to give the horsepower per cylinder.

REFRIGERATION DEPARTMENT

Indicating the Compressor

By Peter Neff

Frequently the question is asked, "Do you advocate the indicating of ammonia compressors? The reply is, Certainly, if it is done right and for some specific purpose. When, however, an engineer mails in a card and asks to have some information regarding his compressor deduced

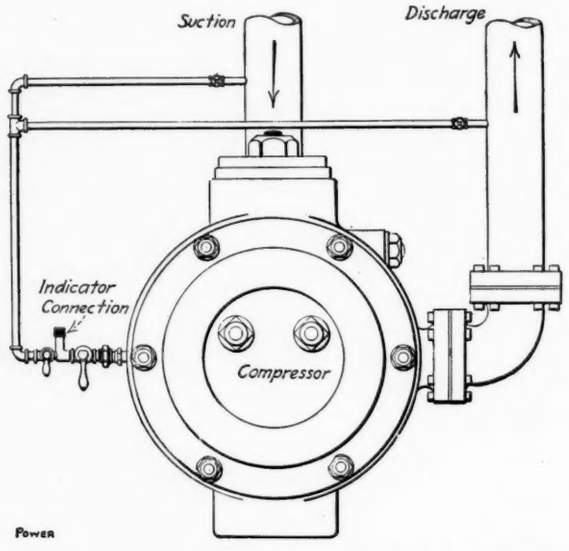


Fig. 1. Indicator Cock Connected to Suction and Discharge Lines

from it, my belief is that any deductions made will be far from being of any value, unless one is perfectly familiar with the engineer who took the diagram, his instrument and methods.

The indicator in the hands of one who has made it a study, and who is proficient in its manipulation is a most useful instrument, but when not so used it may be an erroneous guide, or, to say the least, a slow one.

Usually it is better to trust to an examination of the compressor by a good man, than to what is ordinarily expected to be shown by the indicator. The rythmic noises in an engine room are to the true engineer what an orchestra is to the musician, and he is just as quick to detect when a piece is out of harmony.

The outsider who comes in to inspect a compressor can make more headway by a careful examination than he can by indicating. One must bear in mind that the uses of the indicator for steam engines are not in most cases analogous to those for compressors.

Our best engineers when overhauling their compressor become perfectly familiar with its construction, and the functions which each part performs, and can tell how these should be to obtain maximum results. The piston should be so marked after being adjusted that any movement will be readily noted. No indicator is required for this. A broken valve or spring will manifest itself in many ways to an engineer. And so through the list of most of the troubles to which compressors are liable.

An engineer should have an indicator for his compressor and should know how to use it. It can do much to help him. The point I want to make here is, that many of the things which writers on compressor diagrams show as being determined by the indicator, are as plain as can be to a good engineer without its use.

Some may say that I do not think much of the indicator; far from it, years of study of indicator diagrams from compressors has made me realize, however, that we have in it an instrument with frailties of its own, and much affected by those of man.

In a certain case an engineer spent considerable time in indicating a double-acting compressor, and after much figuring from diagrams taken, made a lengthy report which was in substance that there was too much clearance. It took about one hour to take the head off and determine that there was about $\frac{3}{8}$-in. clearance. He was right, although he did not know the amount, but what a labor to find it out in that way.

For many years there have been published from time to time interesting articles on indicator diagrams, and

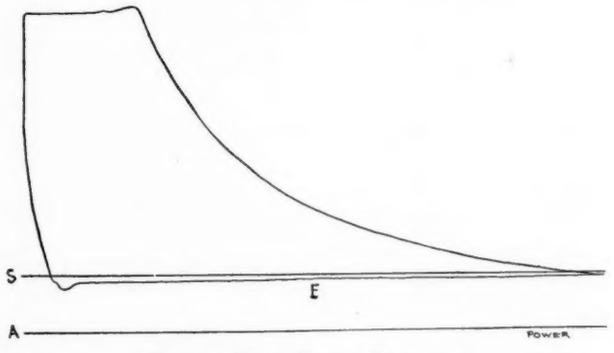


Fig. 2. The Usual Diagram

where these have gone into some great problem connected with compressors, they have been of interest and value. But most of them have merely shown diagrams illustrating various difficulties which could have been known from an examination or other ways more readily.

There have been some published works on the indicator which are splendid and engineers should have them in their possession. The tables and labor-saving devices which they show are helpful, especially in plotting those two useful curves, the adiabatic and the isothermal, which truly are essential to the interpretation of a card.

In compressors, if there is any one thing that we are particularly interested in, it is that all the vapor possible be handled at each stroke. One of the things contributing to this is the filling of the compressor cylinder as nearly as possible to the pressure existing in the suction line. The more nearly this is accomplished the less will be the horsepower and the greater the production of refrigeration. When a 100-lb. spring is employed to

draw the diagram, should the suction pressure be as low as 10 lb., there will be but $\frac{1}{10}$ in. on the diagram in which to study the suction condition. Now to attempt to scale this, especially if a pencil has been used to trace the lines, leaving a comparatively wide mark, is frought with many difficulties even with the use of a magnifying glass.

Very early it was learned to take diagrams for studying suction conditions by the use of a low-tension spring, a 16-lb. spring being a good one for this purpose. Many of the modern indicators are so constructed that these low-tension springs can be used without damage to the instrument. When not so constructed it is usually not a difficult matter to arrange to block the piston so as to prevent any damage arising. With a 16-lb. spring the 10-lb. suction pressure will be quite an appreciable distance on the diagram.

While engine-room gages are all right for general indications, they are by no means accurate. Where there is a mercury-column gage and it has been corrected by the use of a barometer, the pressure can be accurately determined. Unfortunately for most engineers, the engine-room equipment does not contain a mercurial gage, so that usually the readings are taken by ordinary gages. The diagram is scaled to see how it coincides, and in spite of stiff suction-valve springs and small ports it usually comes out that the compressor is filling to suction pressure. Very little attention has been paid to this point, yet there is opportunity for considerable loss, and the indicator can be used to advantage.

In 1907-8, having occasion to make a study of the suction condition in compressors, the author devised a meth-

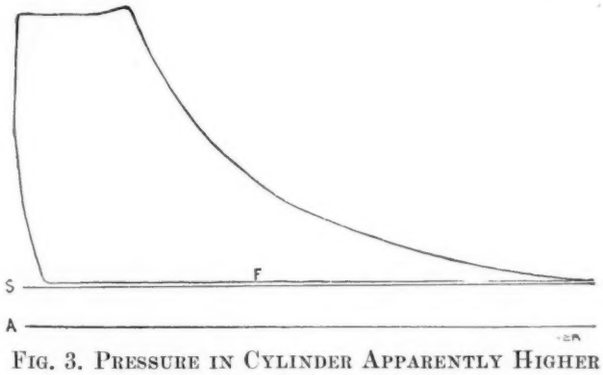


FIG. 3. PRESSURE IN CYLINDER APPARENTLY HIGHER THAN IN SUCTION LINE

od for overcoming this difficulty. If the indicator could in some way be made to record the pressure existing in the suction at the time the diagram is taken, it would manifestly be a guide to go by, which would make unnecessary the scaling of the diagram and the consequent liability to error, as well as doing away with the uncertainty of the engine-room gage reading. In short, the diagram would show to the eye the relative condition of filling pressure to suction pressure.

This condition was accomplished by making a connection from the suction line near the compressor to the indicator, so that when the indicator was shut off from the atmosphere and the compressor, it could be opened to the suction-pressure line. The controlling valve for this should be close to the indicator, as shown in Fig. 1.

It may be urged that as the diagram and the suction-pressure line are not drawn at one and the same instant they are therefore not comparable. A little experience by drawing the suction-pressure line both before and after taking the diagram will show that conditions do not change so rapidly as to materially affect the value of this process. While this method did what was expected of it, it brought out some other interesting facts.

The ordinary diagram obtained would be similar to Fig. 2, in which A is the atmospheric line, S the suction-pressure line, and F the filling line of the diagram. Now the portion of the diagram lying between F and S represents a measurable loss in compressor efficiency, and may be caused by too stiff springs on the suction valves or too small ports for admission of the gas. If the ports are too small the speeding up of the compressor will increase the swing of the line F below S. If the ports are large enough but trouble is with the valve weight or spring, the speeding will not affect it materially.

While indicating compressors where there was practically no drop in pressure for the opening of the valve, I was surprised at times to get a card similar to Fig. 3. Here it will be noticed that the line F is above the line S, in other words, the indicator says that the pressure to which the compressor filled was higher than that existing in the suction line. This may be accounted for as follows:

The tracing of the diagram is in the direction of the arrows, Fig. 4. Now suppose that there is some lost motion in the indicator-pencil connections. The atmospheric line is drawn when the lost motion is all taken up by gravity. As the compression curve is traced the spring is pushing the pencil upward. When the discharge takes place and the pressure begins to decline in the compressor, owing to the friction of the pencil, it is held up while the spring takes up the lost motion, so that the pencil is higher than it should be by the amount of the lost motion. When the suction-pressure line is drawn, the pencil, in the meantime being removed from the paper, falls through the lost motion, and when the suction pressure is applied it is raised by the spring. Both the lines A and S are taken with the lost motion out while F has it.

This led to an examination of indicators and it was found that very few indicators were accurate, and that great care was required in keeping an indicator in fit condition. A slight binding of the indicator plunger will produce the same trouble.

If an indicator had this defect, and a high-tension spring was used for tracing the diagram, and the suction

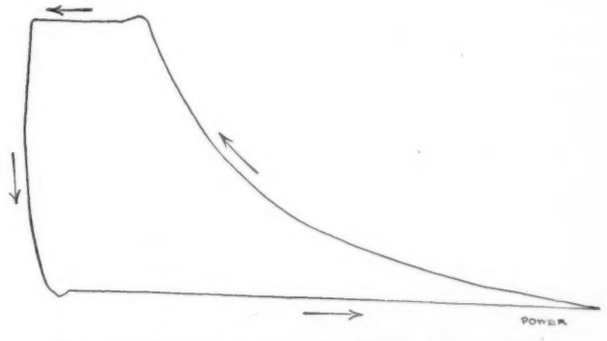


FIG. 4. ACCOUNTING FOR TROUBLE IN FIG. 3

condition determined by scaling the card, manifestly the compressor would be filling to a less amount than shown.

Where there is a considerable drop in the pressure in the compressor before the suction valve will open, this er-

ror will manifest itself in not making the drop as great as it should be, but from that point on the line *F* will be comparable to *A* and *S*.

The fact that this trouble tends to show that the compressor is filling better than it is and further fact that suction pressures are apt to be taken from ordinary gages, has no doubt been a fruitful source of making compressors appear better than they really are.

When studying discharge conditions the effect of lost motion is to considerably distort the graphic record of the discharge. When one adds to the trouble noted, the careless way in which indicators are often connected, it is not to be wondered at that one views with suspicion diagrams submitted for examination. Frequently the cord transmitting the motion from the crosshead does not move parallel with the piston of the compressor, and there is always the uncertain amount of friction of the pencil.

There is one difficulty in drawing the suction-pressure line that must be guarded against. Sometimes liquid will drain into the pipe leading from the suction line to the indicator. When this is the case and it strikes the hot cylinder of the indicator, there will be trouble.

Fig. 5 illustrates another interesting feature brought out in this method of indicating. This diagram has been somewhat distorted to make more plain the point to be illustrated. The suction-pressure line has a loop at both

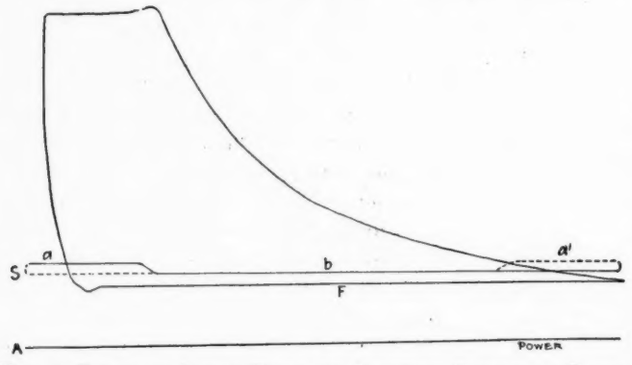


Fig. 5. Loop at Both Ends of Suction-Pressure Line

ends. It must be borne in mind that when tracing either the suction or the discharge-pressure line the conditions existing at both ends of the stroke are shown. In Fig. 5 that portion of the suction pressure corresponding with the diagram is shown as a solid line, while that for the other end for which no diagram is shown is dotted. When the piston is practically at rest at the end of the stroke, the suction pressure indicated by *a* represents the true suction pressure existing in the suction line. Now as the piston moves forward the line *a* falls until it reaches *b*. At the end of the stroke it rises to *a'*. This clearly indicates that the pipe supplying the compressor is not large enough to permit the proper flow of gas. This method has proved interesting in making a study of port velocities, by having an arrangement whereby the area of the port could be changed, and combining this with different speeds.

Fig. 6 is the reproduction of a diagram where not only the suction-pressure line is shown but the discharge pressure as well. Note the banking up of the pressure in the discharge line, and its relation to the discharge of the gas from the compressor as shown by the diagram. This diagram also indicates how difficult it is to study the suction condition when a high-tension spring is used.

A diagram such as this is difficult to obtain, for the use of the indicator with cold suction gas and hot discharge gas is apt to disarrange the working parts to some extent. It is not recommended to try to show all on one diagram. Take up one thing at a time and study that by a series of diagrams.

Should it be desired to reproduce the cards, which is sometimes desirable, an easy method is to put a piece of carbon paper next the indicator drum and use a trans-

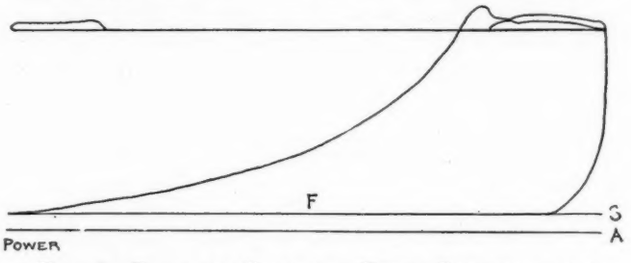


Fig. 6. Diagram Showing Both Suction- and Discharge-Pressure Lines

lucent paper. In this way a dark line may be made on both sides of the paper, enabling a blueprint to be made from it. Care must be exercised as the paper is usually rather thin and a slight excessive pressure on the pencil will cause remarkable distortion sometimes.

My suggestions are to any engineer, to have a good indicator, keep it carefully and do not loan it. Have a good permanent connection for it, including suction and discharge-pressure lines. If possible have a reducing motion permanently connected. If this is not possible, arrange it so that the reducing-motion connections are always the same. Practice the taking of diagrams, for it requires practice to become in any way skilled.

Take up some point for investigation and work it out by a series of diagrams, and these should be plainly marked and dated for future reference. For example, different tension springs can be tried on the suction valves. Or the suction pressure can be taken from different points and the question of the size of the ports and the limit of speed applicable for efficient work determined. Go after the discharge condition. Try to get the compressor to handle a maximum amount of gas with a minimum expenditure of energy.

In doing any of this work many interesting collateral conditions are likely to arise, which can be studied for the benefit of other engineers as well.

In its simplest form refrigeration can be found on practically every vessel afloat, and, according to C. J. Beck, in a paper before the Third International Congress, there are not less than 843 vessels fitted with insulated chambers and refrigerating machinery. The majority of these vessels are thus equipped only for the purpose of carrying fresh food supplies for their passengers and crews, but the cooling equipment is usually quite extensive. A vast fleet of steamships is continually employed in carrying the surplus meats of this country, Canada, Argentina and Australia to the densely populated markets of Europe. The value of the refrigerated ocean transportation to the fruit trade, the fish and dairy industry and many others, is obvious.

EDITORIALS

Saving Our Coal

In these days of coal burning when the enormous demands from land and sea are depleting the supply, it is time that some consideration be given to the fuel of the future. In engineering circles of late years there has been some discussion on this subject and some practical attempts have been made to conserve the coal supply. For a number of years Germany has been utilizing its culm and waste banks in gas producers and has been recovering some of the numerous constituents in coal.

Even this country with its wonderful resources has been awaking to the outlook as evidenced by the developments now under way by the Lehigh Navigation Co., at Hauto, Penn., and those of the Peabody interests in Christian County, Ill. In both it is the intention to generate current at the mouth of the mine and within a reasonable radius supply the demands for power and light. With the lower grades of coal the transportation charge is a large item. When the plant is located at the mine this cost is eliminated and is supplanted by that of transmission which is considerably lower, although the two will finally balance when the distance from the central plant becomes excessive.

These developments, however, are only a beginning. To obtain the last item of value, coal must no longer be treated as a fuel and the raw product burned on the grates for its heat value only. It contains a number of byproducts which in themselves are valuable, and with proper methods of distillation may be recovered. Its treatment should be concentrated at the mines so that the transportation of raw coal may be minimized and the energy transmitted in the form of gas or electricity to the markets. To serve all parts of a large country like our own in this way, where the distances from mine to service are necessarily great, would be impractical at present, but the possibilities for the territories adjacent to the mines, are worth considering.

Along the line proposed above some interesting papers were recently presented in England before the British Association for the Advancement of Science and are to be briefly abstracted in the Oct. 21 issue. Professor Burstall proposes the artificial production of oil fuel. Heretofore coal has been gasified with the object of producing the richest and largest quantity of gas. It is quite practical, however, to change the conditions of carbonization so as to obtain a large yield of fuel oils and other compounds of value and at the same time produce a standard illuminating gas, although this would be a secondary consideration.

As will be seen in the abstract the same author outlined a plan of treatment at the pit's mouth. After separation the tar is subjected to fractional distillation to obtain from it the various oils, and the gas unpurified from sulphur is supplied directly to a gas engine for the generation of electricity. The exhaust gases from the engine are treated in conjunction with liquor from the ammonia plant and the nitrogen and sulphur of the raw fuel recovered in the form of ammonium sulphate.

Professor Armstrong went so far as to suggest that the use of raw coal for heating purposes be rendered illegal. All coal would then be subjected to destructive distillation and the maximum quantity of byproducts would be recovered. The methods of burning the gas would also need reform and Professor Bone's flameless combustion might be expected to receive extended application.

While today these refinements seem visionary, all of them might be practically applied. Utilizing our present resources so as to obtain the maximum value from them is true conservation. It would be in the interest of economy to investigate the plans proposed, for the more fuel we are able to save the more there will be for future generations.

❖

Condenser Tubes

Corrosion and pitting of condenser tubes has long been a perplexing problem in power-plant operation. Many theories have been advanced to account for this phenomena only to be shattered through inconsistent behavior of the tubes under like conditions. Notable among these theories was that of stray currents from outside sources and one company even went so far as to attempt to neutralize these by a counter-current generated within the power house. The experiment failed to prove anything, however, and was discontinued.

Another theory to account for certain tubes deteriorating more rapidly than others in the same condenser and subject to the same conditions, is sagging. It was pointed out that when the condenser is out of service water is allowed to remain in the sagging tubes and hastens galvanic action. Undoubtedly this is a contributing factor, but to what extent is unknown.

The latest investigations while pointing to local galvanic action in the presence of an electrolyte, such as river water containing sulphuric acid, as the primary cause, show, nevertheless, that the extent and rapidity of the action depend upon the tube itself. It is not so much the composition as the structure of the tube which is the governing factor, annealed tubes giving the longest life. Moreover, coarse-grained tubes appear more susceptible to pitting and fine-grain tubes to dezincification.

The beneficial effects of annealing as found by the investigations of the committee appointed by the Association of Edison Companies, reported on page 531, recalls the case of a large power company which came to our attention some time ago. After experiencing much trouble with condenser tubes and trying various compositions, a large shipment was ordered from England and gave complete satisfaction. Upon investigation it was found that these tubes had been made in this country and were shipped to an English firm which put them through a special annealing process. Furthermore, they did not cost any more than tubes bought in this country.

What a License Stands For

One unfortunate effect of an engineers' license is its tendency to swell the head of its possessor. While an engineer may justly take pride in the acquiring of a license, particularly in a state where the examinations are severe, he should not lose sight of the purpose of a license. License laws are passed and licenses distributed, not for the sake of honoring the recipients, but to protect the public.

If an engineer is inclined to be unduly exalted by his license let him remember that barkeepers and peddlers have licenses too, and after all any license is a permit as much as an indorsement—it means that the holder is allowed a privilege, although he has attained a right; he is stamped "good enough" and is not necessarily wonderfully better than his fellows. License examinations have to be easy enough to pass the average as well as hard enough to bar out the incompetent.

Of itself the certificate of a state's approval of an engineer's fitness to perform the duties of his profession, has no virtue. There is no magic about it that can confer ability where none existed before. It does not render him immune from the possibility of making a mistake.

To the really ambitious, conscientious engineer the license is only a milestone indicating that he has reached a certain point in his acquirement of proficiency. He realizes that thus far he has only met the entrance requirements of the race and if he wins a place finally he must not only not relax in his efforts to add to his knowledge, but must double them. The van in the field is easily left, but competition in the forefront is keen.

❋

On the Job

This oft-repeated phrase may mean much or little, depending upon the sense with which it is used and understood. An engineer can be on the job and not accomplish much more than a dog scratching fleas; plenty of action but not getting anywhere.

Day after day and week after week some engineers are on the job. The engines are started each morning on time and are as promptly shut down at night. No accident of a serious nature occurs from year to year and these men are called good engineers.

Look into their plants and the floor and surroundings will be found neat and clean, the engine wiped and the bright parts polished; the engineer is a conscientious man and keeps the plant in the best condition he knows how, but is he *on the job?*

Does he get as much of the heat in the coal into the water in the boiler as is possible? Is the heater raising the feed water to the highest possible temperature, or are the heater tubes covered with grease or partly filled with scale, either of which will prevent efficient work?

The engineer who is *on the job,* not only sees that conditions are right externally, but internally as well, to produce the most economical results. It is not the man who is always in dirty work clothes who is *on the job.* The man who works with his head accomplishes more than he who always has some tool in his hand.

The brains of the successfully operated plant are in the engineer's office, where the schemes for better plant operations are worked out. As a practice, nobody looks to the firemen or oilers for ideas for plant betterment, although both are *on the job* to the extent of their capacity.

More than one chief engineer has failed to make good because he did not have the ability to handle the men in his department. Many owe their success largely to their ability to get their assistants to coöperate with them in the performance of power-plant duties.

If an engineer is *on the job,* he will not only be able to keep the plant running and maintain harmony among the men, but he will be ever on the hunt for losses and for methods of improving the operating conditions.

In one steam plant, operated by a large manufacturing concern, the feed water is pumped from the river through tubes running along the inside of the boiler-furnace setting, these tubes acting as feed-water preheaters. Although the plant has been operated fifteen years, a heater has never been installed because the engines are run condensing and someone decided a heater would be useless.

During these years, however, steam was used in an adjoining building, and a large amount of it has been allowed to escape to the atmosphere. As a result, coal is burned to heat feed water that could have been heated by the escaping steam from the factory building.

Had the engineer been *on the job,* he would have shown the officials of the company that this loss would be prevented by installing a proper feed-water heater.

There are more ways of being *on the job* than always starting up and shutting down promptly and keeping things going meanwhile.

❋

The blowing of an exposed 50-amp. fuse on a motor in a factory room recently created a panic among the fifty or more girl employees, several of whom were injured. This would have been avoided if nonarcing fuses had been used.

❋

Continuously running boilers below rating when fewer would carry the load at or above rating reminds us of the boy who baled water from a sinking boat with a leaky tomato can while a ten-quart pail floated at his feet.

❋

It is only too true that the chief reason the plant owner hesitates to buy needed equipment is because he has been fooled so often that he is afraid to trust or believe anybody.

❋

Some day engineers will learn that the only language the owner understands is so many pounds of steam or kilowatt-hours per dollar and not per pound or ton of coal. Has that day come for you?

❋

We have been informed of a new charge against the isolated plant by central-station promoters. It has been christened "brain fag." No, it has nothing to do with the engineer, but concerns only the plant owner or manager. The reason given for the charge is that the worry, the mental anguish, the "brain fag" which the isolated plant imposes on the manager, is worth so much per year and should, of course, be charged against it. Just how much this charge amounts to per kilowatt-hour or horsepower developed, or whether it has yet been reduced to a unit basis, we are not informed.

READERS WITH SOMETHING TO SAY

Best Results with Producer-Gas Engines

In looking over the Sept. 16 issue, under the heading of "Inquiries of General Interest," there is the following paragraph, being an inquiry of W. G. M.:

"What is the best average result with a producer-gas engine in pounds of coal per brake horsepower-hour?"

The answer below is given:

"About 1½ lb., but most plants use nearer two pounds."

This information is very misleading as the average gas-engine plant uses not to exceed 1¼ lb. of coal per brake horsepower-hour. My record of a plant using approximately 300 hp., for 55 hr. per week, shows for a nine months' run a consumption of 2.13 lb. of coal per kilowatt-hour, this amount including all standby losses during that time.

JOHN H. NORRIS.

Brooklyn, N. Y.

Boiler-Tube Failures

The editorial in the Sept. 2 issue on the above subject is instructive in pointing out the necessity for careful inspection at frequent intervals, particularly with regard to one of the steam user's worst enemies—oil in the feed water.

It would be idle to attempt to enumerate the frequent and well authenticated failures of boiler tubes and sheets resulting from oil being fed into boilers, even in small and apparently insignificant amounts. So long as careless engineers imagine that any return that has had the coarse or floating grease removed is good enough to use as boiler feed without further treatment, just so long shall we continue to read of cases of negligence similar to the one mentioned and, until it is indelibly impressed upon the minds of easy-going engineers that a cloudy or even slightly turbid-looking return water contains dangerous quantities of oil, just so long will the lives of firemen and others be in danger, and the property of manufacturers liable to destruction.

The editorial clearly indicated the cause of the tube failure in the instance mentioned: First, the oil separators in the steam line had not been operating effectively. Second, apparently no filter was used at all, to say nothing of a good filter as the editorial suggested.

This latter point is really the more important, as a good filter compensates for the deficiencies of the separators which are unreliable in separating the finely divided and vaporized oil.

A large majority of separators give satisfactory results only for a short time after being installed because they soon coat with a film of oil which is readily picked up again by the current of steam and carried to the feed-water heater as emulsified oil. Hence, the necessity of a filter—and a good filter.

Inasmuch as in most installations the water gets no further treatment to remove the oil, the filter should be of such character that it will absolutely remove all oil, both coarse and emulsified. If the water after passing through such filter looks hazy, cloudy or even only slightly milky, the filter should not be accepted or installed. Condensation from returns should look perfectly clear to be safely satisfactory. The use of alum or other chemicals is undesirable as constant attention is required to prevent any possibility of acids or other corroding substances getting into the boilers or eating out the valves and connections.

The importance of completely removing grease or oil from the feed water may be more fully appreciated when it is remembered that grease offers ten times more resistance to the passage of heat than ordinary boiler scale, and that the merest trace of grease, less than $^1/_{1000}$ of an inch, can cause far more serious injury than scale.

The dangers due to the presence of grease, etc., should not discourage the use of grease-impregnated water, provided an efficient separation of the oil is effected. This can now readily be done by several means, one of which is a system of absorbtion filtration, the filtration medium being a silicate fiber with strong physical attraction or affinity for oil and grease.

A. E. KRAUSE.

Jersey City, N. J.

Orsat or CO_2 Recorder Best?

Mr. Pohlman's argument, expressed in his letter under the above title in the July 29 issue, that CO_2 recorders are easily broken by inexperienced hands does not seem to apply to some of the more simple modern types.

As regards the relative value of the continuous recorder and the Orsat type, occasion may require that both should be used, for purposes of comparison.

Assume, for argument, that a given fuel burned with a minimum air should show 20 per cent. CO_2 and that the actual reading shown was 19 per cent. It is probable that the other 1 per cent. was manifested in the form of CO, showing imperfect combustion. No engineer would try to better such results. But if only 6 per cent. of CO_2 were shown, nobody would be likely to think the balance made up of carbon monoxide for, clearly, great excess air is indicated. And so, in practice, if the CO_2 is recorded above 12 per cent. the tendency is to leave well enough alone. There may or may not be a small amount of CO present, and if more air were added to correct this the overall efficiency would undoubtedly suffer. Again, if the CO_2 is in the neighborhood of, say, 9 per cent. and CO in appreciable quantity is present, effort should be made to improve the result. But the improvement would not be accomplished by admitting more air but rather by altering the furnace arrangements. Carbon monoxide is not largely

produced in boiler furnaces. When the air supplied is greatly in excess of the chemical requirements it is unlikely to be formed in a properly arranged furnace. When the CO_2 is high the monoxide even if present cannot be reduced. This fact favors the continuous recorder.

An occasional check test should be made with the Orsat apparatus. Personally, the Orsat does not appeal to me as of much use unless worked on the long-sample method, a sample taken over a long period of time. This system shows only average conditions while the recorder gives a diagram which not only can be measured up for averages but shows at very numerous intervals the varying conditions that make up the averages. And so far, the recorder is of greater value for we do not merely require averages; we want to know why the average does not approach more nearly to the best.

The Orsat apparatus should be used along with the recorder, quick samples being taken at maximum and minimum points of the recorder diagram so that the percentage of CO and of free oxygen can be found. Wherever these two gases are present together the inference must be that the mixture has been incomplete up to the time when the gases became too cold to combine. Carbon monoxide is fairly greedy for oxygen and could not remain hot in contact with it without conversion. Carbon monoxide is thus indicative of faulty mixture. It rarely happens that in boiler furnaces there is not more than enough oxygen for all purposes if only it is well distributed.

W. H. Booth.

London, Eng.

Water in the Engine Cylinder

The article in the Aug. 26 issue describing the tests made by H. Bollinckx as to water being trapped in the cylinder and doing damage is of more than passing interest inasmuch as it disproves the theory that a large percentage of the engine cylinders that are bursted are caused by entrapped water.

H. R. Low's article in the June 25, 1912, issue shows how the water may get into the low-pressure cylinder, but Mr. Low does not say anything about its getting out, and if it came from the boilers it would have practically the same chance to do damage in the high-pressure as in the low-pressure cylinder.

The tests made by Mr. Bollinckx were of such nature that if water in the cylinder could have ruptured it, it would have done so in some of the tests that he made. This would apply only to cylinders in which the exhaust valve is placed in the bottom of the cylinder, as the greater portion of the water would lie on the bottom of the cylinder, and it could not escape as readily from a cylinder on which the valve was placed on the side or top of the cylinder. In the latter case a small amount of water would very likely cause a damaged cylinder.

Every engine ought to be equipped with automatic relief cocks which would allow water to escape that should be trapped in the clearance space no matter from what source it came. Furthermore, every precaution should be taken to keep the water from going into the cylinder as it will make lubrication more difficult, even though it does no damage. An efficient steam separator should be placed in the steam line as near the cylinder as possible, and connected to a good reliable trap or steam loop, and also a large drip pocket should be placed in the exhaust line with a trap or pump connected to it to remove the condensation.

I have in mind a 30x48-in. long-range cutoff Corliss engine running at 90 r.p.m. with a variable load that pulled much water into the cylinder, due to a long steam line. There was a separator near the engine with an open drip, but at times the 2½-in. relief cocks would blow a solid stream of water at each stroke for two or three minutes at a time. In this case the water did not seem to get out of the cylinder fast enough, but a part of it was trapped in the compression space after the exhaust valve closed, and was compressed high enough to open the relief valves which were set at 20 lb. above the highest steam pressure to prevent excessive strain on the cylinder. The exhaust pipe led direct to the atmosphere with a 1½-in. drip pipe tapped into the lowest point which was left wide open all the time.

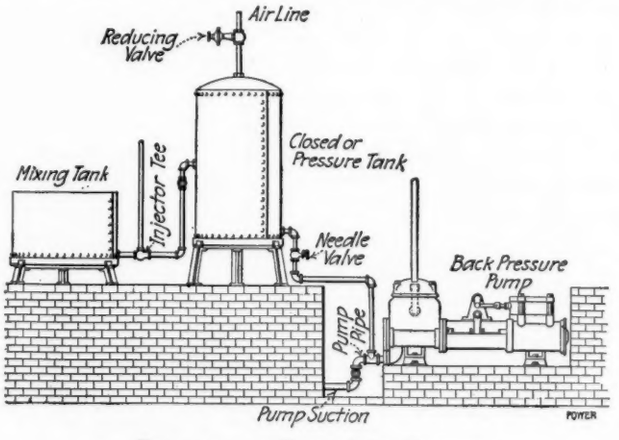
Engineers would do well to re-read these two articles as they deal directly with everyday troubles, and incidentally we will have to find some other excuse for the not infrequently bursted cylinder heads, pistons and bent piston rods that in the past have been charged up to water in the cylinder as has been done in the past for want of a better reason.

J. C. Hawkins.

Hyattsville, Md.

Feeder for Soda Ash

Readers may be interested in knowing of my experience with a compound-feeding device, similar to that described by Mr. Anderson, in the Aug. 12 issue. Having found that plain soda ash, which may be purchased for 1½c. per lb., was especially adapted for treating the



Piping for Soda-Ash Feeder

boiler water used in our plant, I decided that the best results could be obtained when this chemical was fed continuously instead of spasmodically. I, therefore, rigged up a tank for gravity feed, somewhat like the tank shown by Mr. Anderson, although this tank did not contain heating pipes. As the headroom was very small, I found that it was difficult to inject the solution into the feed-pump suction pipe, so a pressure system was installed, as shown in the illustration.

Sufficient chemical solution for the run is mixed in the open mixing tank, and then transferred to the closed or

pressure tank by means of a steam ejector. The pressure tank is then closed and compressed air is admitted to maintain a constant pressure of 5 lb. through a reducing valve. This gives a constant head on the entire system and enables a close adjustment of the rate at which the solution is fed through the small needle valve into the feed-water pipe. The feed is adjusted so that the boiler water is made slightly alkaline. With this system, irregular feeding, due to changing of the solution, is eliminated. I have also found that under this slight pressure, less trouble is experienced with clogging of the needle valve.

JOHN SMITH.

Philadelphia, Penn.

Test Results for 24-Hour Run

This is a copy of the report of our average run. I trust others will submit similar reports so that we, as engineers, may be given an opportunity to compare the results we get from our plants.

DAILY LOG SHEET—AUGUST 29, 1913

Plant	C. P & Co.
Location	Waterford, N. Y.
Boiler heating surface, sq.ft.	5524
Rated horsepower	552
Type of furnace	Universal stoker
Grate surface, sq.ft.	100
Fuel	Bit. run-of-mine
Trade name	Cascade
Duration of run, hr.	24
Steam pressure by gage, lb.	145
Temperature of feed water, deg. F.	267
Degrees of superheat	
Factor of evaporation	0.9919
Blast under grates	
Draft in furnace, in. water	0.04
Draft at dampers, in. water	0.42
Temp. of escaping gases, deg. F.	500
Total water fed to boilers by meter, lb.	675,503
Total water fed to boilers less temp. corrections of 15 per cent., lb.	574,178
Equivalent evaporation from and at 212 deg. F., lb.	569,494
Equivalent evaporation from and at 212 deg. F. per hr., lb.	23,728
Equivalent evaporation from and at 212 deg. F. per sq.ft. heating surface per hour, lb.	4.29
Horsepower developed	687.7
Per cent. of rated horsepower developed	124.5
Total fuel fired, lb.	50,500
Moisture in fuel, per cent.	1
Total dry coal, lb.	49,950
Total combustible, lb.	45,100
Dry fuel per sq.ft. grate surface per hr., lb.	20.81
Refuse by weight, per cent.	10.6
CO_2, per cent.	12
Flue-gas analysis CO, per cent.	8
Flue-gas analysis O, per cent.	0.6
Volatile matter, per cent.	18.29
Approx. analysis, fixed carbon, per cent.	72.78
Ash, per cent.	7.92
B.t.u. per lb. fuel	14,529
Efficiency of boiler and furnace per cent.	
2 boilers running without an economizer	

C. R. WARD.

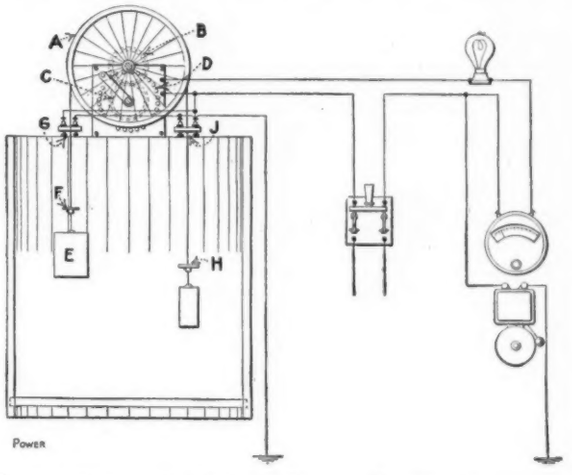
Waterford, N. Y.

Voltmeter Indicates Water Level

The plant I have charge of has a standpipe located on top of a hill a mile away, and it has always been a difficult matter to tell the exact amount of water in the standpipe by the pressure gage at the plant, for when the pump is running fast the gage registers more water than is in the standpipe. When the pump is stopped and great quantities of water are being drawn from the mains, the pressure gage will register less than is in the standpipe. We have arranged a rheostat on the standpipe roof, which raises and lowers the voltage on the voltmeter that is located at the plant, as the water rises and falls. The voltmeter is calibrated in feet, so it gives the direct reading of the water in the standpipe.

Referring to the illustration, *A* is a bicycle wheel mounted on a frame on the roof of the standpipe; *B* is a small gear fastened to the hub of the wheel, which meshes in the gear *C* to which is attached the rheostat-operating lever; *D* is the rheostat. As the water rises, the float *E* turns the wheel, which cuts out resistance in *D* and causes the voltage to register on the voltmeter. As the water falls it cuts in resistance and lowers the voltage, as indicated by the meter. The current is supplied to the circuit, as shown in the drawing.

When the float rises within two feet of the top of the standpipe the metal ring *F* closes the contacts *G* and



RHEOSTAT WATER-LEVEL INDICATOR

rings the bell at the plant. Then when the float gets within two feet of the bottom metal ring *H* closes the contacts *J* and rings the bell.

A. G. VINSON.

Yazoo City, Miss.

Success of Graphite in Boilers

I have read the letters from E. C. Sutton, of Estherville, and Daniel I. Curtin, of North White Plains, N. Y., telling of their experience with graphite.

I am employed by the Armstrong Cement Works, of Armstrong, Iowa, and have charge of the power plant. I have used soda ash in the boiler with success, but when we installed a new boiler, feed-water heater and pump, we had trouble with the pump clogging so decided to try graphite. Before we used graphite there was a soft substance in the pump which could easily be washed out with a strong stream of water, now since the use of graphite there is a scale about ⅛ in. thick coated with graphite which adheres to the valves and springs, which keeps them from working and is also very hard to remove. I have been using my pocket knife to scrape it off.

The last time I cleaned the pump I mixed a strong solution of soda ash and put it in the pump, leaving it there about two hours, this made it easier to remove. To keep the pump in good working order it has to be cleaned every two or three weeks. The heater is cleaned once a month. There is a scale on the pans that is usually 1¼ in. thick.

From the water line down half way on the sides of the boiler the shell is coated with graphite with no scale under it but the lower half has a thin scale about $\frac{1}{32}$ in. thick with graphite. The tubes are in the same condition.

We have a graphite feeder connected with the suction pipe of the pump in which the graphite is fed continuously.

We have used graphite about sixty days. The first two weeks we used it according to directions then we doubled the dose with no better success. Before we started using graphite there was no scale in the boiler. We have some graphite left and have decided to try graphite and soda ash mixed.

Test of Valve and Piston Tightness

On page 121 of the July 22 issue is a description of a test, as above, of a Ball engine. One brief remark strikes me as being worthy of educational note—"it is plain that no steam is escaping, although it is reported that a few drops of water were discharged intermittently."

Here is a practical demonstration of one of the greatest benefits which is to be derived from the use of superheated steam—no leaks past the piston and valve-gear. When the steam is wet the moisture in it can get through the most restricted passage, but the steam itself will not escape. The steam tightness of the valves and piston of the Ball engine as described was perfect, and yet moisture could get through. How much more than "a few drops intermittently" would come past the valves and piston of an engine whose steam tightness is such as is commonly found in practice, may better be imagined than described. In a cylinder of any size it would not take long to fill a bucket.

The obvious moral is superheat if practicable, but unless there are other outside conditions dictating high degrees of superheat, remember that there is no economy in carrying the superheat to a higher degree than will enable the steam to be dry all the time it is in the engine cylinder.

JOHN S. LEESE.

Manchester, Eng.

Why Trap Failed to Drain Cans

L. Johnson asks a very interesting and instructive question in the issue of Sept. 9. It is to be regretted that he did not submit a sketch with it or give more definite information, as in its present state many things are unknown to men of much experience who would gladly help solve the question.

Was the trap a return or separating trap? Assuming it was of the return variety, was it installed in the first place beyond the receiver, or in the same relative position as the pump that it was installed to replace, and did replace later on? And, again, how much, if any, was the top of the bowl or tank of the trap when in its filling position below the outlet of the cans?

Without definite information, it would seem as if the top of the trap as first installed was very near the same level as the outlet of the cans, or possibly a trifle above it, so there was not enough head to fill as fast as necessary to keep the cans free, for it must be remembered that under certain conditions it takes a trap as long to fill as it does to empty. A pumping, lifting, or return trap, is only delivering its contents one-half the time, and it seems as if in this case the trap filled so slowly that while it was discharging the water accumulated in the cans, and when it was removed to its second position close to the cans it was set somewhat lower, or if on the same level then the mere difference in friction of the supply pipe was enough to remove the cause of disorder.

W. H. ODELL.

Yonkers, N. Y.

Preventing Freezing of Air-Driven Pumps

In reply to H. R. Leonard's question, on page 138, July 22 issue, in regard to the freezing of the exhaust of an air-driven pump, I would say that the freezing is due to the presence of moisture in the air.

In a number of plants where pumps, rock drills, etc., are operated by compressed air, the moisture is successfully removed from the compressed air to obviate this difficulty. It is well known that the capacity of air to carry water vapor increases as the temperature rises, and diminishes as the temperature is lowered. What is known as "aftercoolers" are frequently used for this purpose. These are generally built like surface condensers, or coils of pipe, in which the air after leaving the compressor is cooled as low as possible by cool circulating water, after which the air is passed through a receiver or separator to remove the water that is condensed from the air. If this is properly done there seldom is much trouble from freezing. Sometimes a separator or receiver is used near the place where the air is used as the air is generally cooled down enough to precipitate most of the moisture if it is any considerable distance from the compressor.

On sinking pumps, etc., in mines a small stream of water is often allowed to drip back over the pump.

G. A. REICHARD.

Los Angeles, Calif.

Stops Leak with Babbitt Metal

While in charge of a small steam plant, I was troubled with a leaky steam pipe. The leak occurred at the screw joint, which I had several times attempted to stop by screwing the coupling up tighter, by calking, and with cements, but the steam continued to escape. The pipe was so old and rusty that I could not tighten it any more, and I could not make adhesives stick permanently.

The pipe was continued from the boiler to an engine, a distance of about 50 ft. The leaky joint was just under the floor of the engine room where there was very little space for working, but, inasmuch as the joint was not visible, I decided that even a poor-looking job, if thorough, would be satisfactory.

It occurred to me that a mold could be formed around the pipe with little trouble and that if the mold were filled with hot babbitt completely around the leak, there would be no more trouble. I, therefore, bent a strip of sheet metal around the pipe at the troublesome spot, bent up the ends, puttied up the sides to render the mold leakless and poured the babbitt. Although the finished job does not look first-class, the leak no longer exists.

The method may appeal to some engineers as a means for effecting good, temporary repairs as well as permanent ones.

S. F. WILSON.

Brooklyn, N. Y.

[It would have been far better to renew the corroded pipe instead of stopping the leak as described.—EDITOR.]

INQUIRIES OF GENERAL INTEREST

Height for Glass Gage—At what height should a glass water gage be placed on a horizontal tubular boiler?

W. H.

At such height that the lowest visible part of the glass will be at least 2 in. higher than the top of the highest tube of the boiler.

Laying Up a Storage Battery—If a storage battery is to be put out of service for an indefinite time, what attention should it receive?

A. A. T.

Charge the battery to its full capacity and then remove the acid and fill the cells with distilled water, and when again required, refill with acid, and give the battery an overcharge; or discharge the battery until the voltage is practically zero, and after rinsing with distilled water, allow the plates to dry.

Radius of Bumped Head—What would be the radius of a convex head which is 30 in. dia. and bumped 4 in.?

R. A. S.

Calling the radius R, then

$$R^2 = (\tfrac{30}{2})^2 + (R - 4)^2$$

or

$$R^2 = 225 + R^2 - 8R + 16$$

from which

$$8R = 241 \text{ or } R, \text{ the radius } = \tfrac{241}{8} = 30\tfrac{1}{8} \text{ in.}$$

Horsepower of A.C. Motor—How is the horsepower found on a three-phase a.c. motor when voltage and amperage are given?

E. F. R.

The power of a three-phase motor is given by the formula

$$KW = \frac{\text{Current} \times \text{voltage} \times 1.73 \times \text{power factor}}{1000}$$

This applies to the current in any phase, and the voltage is that measured across any two of the live wires. As one horsepower is equivalent to 0.746 kilowatt, the horsepower of the motor would be the kilowatts found by the formula divided by 0.746.

Running Motor as a Dynamo—Will it be possible to run a 100-kw. direct-current motor as a dynamo? What changes, if any, will have to be made?

E. P. B.

Any direct-current motor can be used as a dynamo. The only difference is that when used as a motor, it has a low-resistance rheostat in series with the armature; whereas, when used as a dynamo, it is provided with a high-resistance rheostat, in series with the shunt field.

If the machine is compound-wound it will be necessary to interchange the two terminals of the series field winding to prevent it from opposing the shunt field winding, also, it will be necessary to speed up the machine slightly above the speed at which it ran as a motor, to compensate for the voltage drop in the armature circuit.

Variation in Pressure Along Pipe Line—When water from a pump is being discharged through a horizontal pipe line under pressure, is the pressure the same at all points along the pipe line?

M. N.

The pressure will not be the same; beginning at the pump, friction will cause the pressure to drop from point to point all along the line.

Heat Absorbed by Ice—How much heat will be absorbed by a block of ice 4x6x4 ft., at 32 deg. F. in becoming melted to water?

H. E. H.

Average ice weighs about 57.5 lb. per cu.ft. and a block of the size mentioned would contain 96 cu.ft. and weigh 5520 lb. The latent heat, i.e., the heat required to melt a pound of ice at 32 deg. F. to water being 144 B.t.u. the heat required would be $5520 \times 144 = 794{,}880$ B.t.u.

Steam Discharge from Pipe—What quantity of steam would be discharged per hour by a ¾-in. pipe kept filled with steam at 125 lb. boiler pressure, the open end of the pipe being submerged 1 ft. below the surface of water in a tank?

S. S. G.

The discharge would be practically the same as would be given by Napier's approximate formula for the outflow of steam into the atmosphere viz.: Pounds of steam per second $= \frac{pa}{70}$, in which p = absolute pressure in pounds per sq.in. and a = area of orifice in square inches. The absolute pressure being 125 + 14.7 = 139.7 lb. per sq.in. and as a ¾-in. dia. orifice would have an area of 0.44179 sq.in., then according to the formula the discharge would be

$$\frac{139.7 \times 0.44179}{70} = 0.8816 \text{ lb. per second, or}$$

$$0.8816 \times 60 \times 60 = 3173.76 \text{ lb. discharged per hour.}$$

Pressure After Cutting Off—With an engine having 7 per cent. clearance and cutting off at half stroke, what would be the pressure at three-fourths stroke?

D. T.

The absolute pressure at three-fourths stroke would be

$$\frac{0.5 + 0.07}{0.75 + 0.07} = \frac{0.57}{0.82}$$

or about 69½ per cent. of the absolute pressure at cut off.

Gasoline—What are the different grades of gasoline, and is there any difference in weight per quart; also how many B.t.u. are there per pound? What amount of gasoline would be consumed by an engine per hour?

J. M. H.

There is no distinct division in the grades of gasoline, it being designated according to its specific gravity, which varies from 65 to 74 Baumé scale. The heat value varies with the specific gravity from 17,000 to 19,000 B.t.u. per pound. For the four-cycle engine, under ordinary conditions, it is customary to assume a consumption of one pint of gasoline per horsepower-hour.

Measuring Temperature by Specific Heat—What is the method of determining the temperature of a furnace fire, or of flue gases by heating and cooling a piece of fire brick or other substance?

W. G. G.

A given weight of the fire brick or other substance of known specific heat, is placed in the furnace fire or flue gas whose temperature is required and after becoming heated to the unknown temperature, is plunged into a known weight of water and the rise in temperature of the water is noted. If

X = The unknown temperature of furnace or flue gas;
w = Weight of heated substance, lb.;
W = Weight of cooling water, lb.;
T = Final temperature of the water;
t = The temperature rise, or difference between initial and final temperatures of the water;
s = The specific heat of the cooled body;

the heat lost by the cooled body being equal to the heat gained by the water, then

$$ws \times (X - T) = W \times t$$

i.e.,

$$X = \frac{Wt}{ws} + T$$

The mean specific heat of fire brick for high temperatures is about 0.26 and if a piece of fire brick weighing 3 lb. were to be placed in a furnace, and heated to the temperature of the fire and then, without loss or gain of heat, is placed in a bucket containing 18 lb. of water of original temperature of 60 deg. F.; and after the immersion of the heated substance and without other loss or gain of heat, the temperature of the water rose to 150 deg. F., then the furnace temperature would be:

$$X = \frac{18 \times 90}{3 \times 0.26} + 150 = 2227 \text{ deg. F.}$$

ENGINEERS' STUDY COURSE

Heat Balance

In preceding lectures a method was given for conducting a boiler test, making and estimating the efficiency and economic results. After a test has been made the owner or engineer may be dissatisfied with the showing made. He may have been justified in expecting a better efficiency when the design, arrangement and condition of the boiler, furnace setting, etc., are taken into consideration. The test report alone does not show him in what particular the operation was poor, it simply gives final results. To get fuller information as to how the operation can be improved it is necessary to analyze the distribution of the heat contained in the coal. Perhaps in every pound of coal thrown into the furnace there was 14,000 B.t.u. available. If all this heat could be put into the water, an efficiency of 100 per cent. would be obtainable.

Unfortunately, this is impossible—some loss is bound to take place. At the very beginning, some of the burnable matter falls through the grate with the ash. Hence, all the combustible matter of the original pound of coal is not burned and, consequently, the full 14,000 B.t.u. is not generated. Next, the moisture in the coal must be evaporated and superheated and all the heat that such moisture carries out with the flue gases is lost.

The air fed into the furnace all passes out again to the chimney. True, a part of it has undergone a change (the oxygen combining with the hydrogen and carbon of the fuel) but it is all there nevertheless. This air leaves the boiler at a considerably higher temperature than that at which it entered. The heat added to it was obtained from the coal—and nowhere else. Hence, here is another loss. And so it goes; some heat radiates from the boiler and setting, and some more heat is lost through the incomplete combustion of the carbon in the fuel. Some heat is lost through unburned particles of combustible matter passing up the stack as soot, and some is lost because all the heat created by the combustion of the hydrogen in the fuel cannot be recovered in the boiler.

An itemized statement in B.t.u. and percentage of all these losses together with the quantity of heat absorbed by the water in the boiler to make steam is called a heat balance. To itemize each loss individually would necessitate considerable trouble in the way of obtaining the required data so only six divisions are usually made:

(a) Loss due to moisture and hydrogen in coal.
(b) Loss due to heat carried away by dry flue gases.
(c) Loss due to incomplete combustion.
(d) Loss due to coal dropping through grates.
(e) Heat unaccounted for (this item includes all losses not taken into consideration in the four foregoing items).
(f) Heat put into the water.

Item (a) is made up of two factors: 1, the loss of heat required to change the moisture in the coal into steam at the temperature of the gases leaving the boiler and, 2, the loss due to the fact that all of the heat generated by the combustion of the hydrogen of the fuel cannot be used for making steam in the boiler. When hydrogen burns it forms water in the shape of highly superheated steam which in the fuel calorimeter is condensed so that the latent heat of this steam passes into the cooling water surrounding the bomb (see lesson in June 10 issue) and is credited to the coal. In actual use in the boiler, however, the steam formed by the combustion of the hydrogen is partly cooled by contact with the heating surface of the boiler, but it is not cooled sufficiently to condense and, consequently, it passes off in a superheated condition at the temperature of the chimney gases. Hence, the additional heat it would give up if cooled down to ordinary temperatures and condensed is lost.

The heat given up when a pound of superheated steam at atmospheric pressure is condensed and cooled from 500 deg. to 70 deg. is exactly equal to the heat required to evaporate and superheat a pound of water from 70 to 500 deg. Hence, for simplicity and convenience it is customary to calculate the amount of water that would be formed from the estimated quantity of hydrogen in the coal and add this to the amount of moisture in the coal as shown by the proximate analysis and then calculate the heat required to raise the total quantity to the boiling point, convert it into steam and superheat it to the temperature of the escaping gases.

The first step, then, is to estimate the percentage of hydrogen in the coal, for it is seldom that an ultimate analysis is available in commercial practice. The following formula, originated by Professor Diederichs and published in POWER of Jan. 14, gives the approximate total hydrogen in the combustible matter:

$$H = v\left(\frac{7.35}{v + 10} - 0.013\right)$$

where

H = Percentage by weight of hydrogen in the combustible matter;

v = Percentage by weight of volatile matter in the combustible matter.

For illustrating the application of this formula and other calculations required in constructing a heat balance the data and results of the imaginary test reported in the Sept. 30 lesson will be used. In the calculations last week it was found that the percentage of volatile matter in the combustible was 4.37. Substituting in the foregoing formula we have

$$H = 4.37\left(\frac{7.35}{4.37 + 10} - 0.013\right) = 2.18 \textit{ per cent.}$$

hydrogen in the combustible matter. As there is only 79.84 per cent., or 0.7984 lb. combustible in each pound of coal as fired, the percentage of hydrogen in the coal as fired is only

$$(2.18 \times 79.84) \div 100 = 1.74 \textit{ per cent.}$$

or 0.0174 lb. hydrogen per pound of coal as fired.

In the lesson of May 6, it was shown that the combustion of every pound of hydrogen results in the production

of 9 lb. of water. Consequently, the combustion of 0.0174 lb. of hydrogen results in the production of

$$0.0174 \times 9 = 0.157 \text{ lb.}$$

The proximate analysis of the coal used in the test showed that in each pound of coal as fired there was 0.07 lb. of water. Then, the loss due to the hydrogen and moisture in the coal as fired is equal to the heat required to evaporate and superheat

$$0.157 + 0.07 = 0.227 \text{ lb.}$$

of water from 92 to 540 deg., the temperatures given in the test report.

The heat required to raise this quantity of water from 92 deg. to the boiling point, 212 deg., is

$$(212 - 92) \times 0.227 = 27.24 \text{ B.t.u.}$$

The heat required to change this water into steam from and at 212 deg. (latent heat) is

$$970.4 \times 0.227 = 220.28 \text{ B.t.u.}$$

And, assuming the specific heat of superheated steam to average 0.46, the heat required to superheat the steam from 212 to 540 deg. is

$$(540 - 212) \times 0.227 \times 0.46 = 34.25 \text{ B.t.u.}$$

Hence, the total loss per pound of coal as fired, due to the hydrogen and moisture, is

$$27.24 + 220.28 + 34.25 = 282 \text{ B.t.u.}$$

As the heat value of the coal as fired was 11,718 B.t.u. per lb. this loss expressed in percentage is

$$(282 \div 11{,}718) \times 100 = 2.4 \text{ per cent.}$$

In the present case this loss is not very high, because the coal used was of the anthracite grade in which the percentage of moisture and hydrogen never runs very high. But, with other grades, especially the bituminous and lignite, the loss runs considerably higher.

Item (b) of the heat balance is the most important because it is the largest and its magnitude depends greatly upon the skill and care used in operation.

A formula for estimating the heat lost up the chimney per pound of coal burned was given in the July 22 lesson as follows:

$$L = 0.24\ W\ (T - t)$$

where,

L = B.t.u. lost up the chimney per pound of fuel burned;

W = Weight of flue gas formed per pound of fuel burned;

T = Temperature of the gases leaving the boiler;

t = Temperature of the air entering the furnace.

The weight of the flue gases formed per pound of coal burned is estimated by another formula in the July 22 issue as follows:

$$W = 3.032\ C\left(\frac{N}{CO_2 + CO}\right) + (1 - A)$$

where

C = Weight of total carbon in the fuel as fired;

N, CO_2 and CO = Percentage by volume of nitrogen, carbon dioxide and carbon monoxide, respectively, in the flue gases;

A = Weight of ash in the fuel as fired.

When the volatile matter in the combustible amounts to 12 per cent. or more the total carbon in the coal as fired is found with the aid of the chart in the July 22 issue. Where it runs less, it is safe enough to assume that the volatile carbon is approximately one-third of the volatile matter. Thus, in the present case, the volatile carbon in the coal as fired would be

$$3.49 \div 3 = 1.16 \text{ per cent.}$$

and the total carbon would be, therefore,

$$76.35 + 1.16 = 77.51 \text{ per cent.}$$

or 0.7751 lb. per lb. of coal as fired. Substituting in the foregoing formula this value and the values for the flue gases as given in the test report we have:

$$W = 3.032 \times 0.7751 \left(\frac{80.2}{10.7 + 0.4}\right) + (1 - 0.1316)$$

$$= 17.85 \text{ lb.}$$

And applying this factor in the formula for heat loss we have

$$L = 0.24 \times 17.86\ (540 - 92) = 1920 \text{ B.t.u.}$$

per pound of coal burned, carried up the chimney by the flue gases. Expressed in percentage this equals

$$(1920 \div 11{,}718) \times 100 = 16.38 \text{ per cent.}$$

The method of calculating item (c) was given in the Aug. 5 lesson. The formula is,

$$L' = 10{,}150\left(\frac{CO}{CO + CO_2}\right) c$$

in which L' equals the heat loss due to incomplete combustion per pound of coal burned and expressed in B.t.u., and the other symbols represent the same values as before. The heat loss in the present case is

$$L' = 10{,}150\left(\frac{0.4}{0.4 + 10.7}\right) 0.7751 = 283 \text{ B.t.u.}$$

or

$$(283 \div 11{,}718) \times 100 = 2.41 \text{ per cent.}$$

In last week's lesson it was estimated that there was 532 lb. of combustible matter in the ash and refuse. As the heat value of the combustible was found to be 14,677 B.t.u. per lb., the total heat loss due to unburned coal dropping through the grates was

$$14{,}677 \times 532 = 7{,}808{,}164 \text{ B.t.u.}$$

And, consequently, the loss per pound of coal fired (item (d) of the heat balance) was

$$7{,}808{,}164 \div 10{,}372 = 753 \text{ B.t.u.}$$

which, expressed in percentage equals,

$$(753 \div 11{,}718) \times 100 = 6.43 \text{ per cent.}$$

Item (e) is the difference between the heat value of the cal as fired and the sum of all the other items.

The efficincy of the boiler and grate was found to be per cent. In other words,

$$(11{,}718 \times 68.98) \div 100 = 8083 \text{ B.t.u.}$$

was put into the water in the boiler to make steam. Then, item (e), representing all other losses or losses unaccounted for must be,

$$11{,}718 - (282 + 1920 + 283 + 753 + 8080) = 397 \text{ B.t.u.}$$

$$(397 \div 11{,}718) \times 100 = 3.39 \text{ per cent.}$$

All the items of a heat balance are usually arranged in tabular form, thus:

HEAT BALANCE

Heat value of 1 lb. coal as fired = 11,718 B.t.u.

Distribution of Heat	B.t.u.	Per Cent.
(a) Loss due to moisture and hydrogen in coal	282	2.41
(b) Heat carried away by flue gases	1,920	16.38
(c) Loss due to incomplete combustion	283	2.41
(d) Loss due to unburned coal dropping through grates	753	6.43
(e) Losses unaccounted for	397	3.39
(f) Heat absorbed by boiler	8,080	68.98
Total	11,718	100.

OVER THE SPILLWAY

JUST JESTS, JABS, JOSHES AND JUMBLES

Old Sol, to whom we have always looked as an inspirer of warmth, happiness and good cheer, is in bad out Kansas way. He's been a veritable firebug during the hot spell. He'd hit a tin roof, and blim! the house burned up; he'd focus on a barb-wire and blam! a hayfield was consumed; when gathered by a lantern globe, the ancestral home was doomed before the vamps could find their fire hats. Solly, we're a bit disappointed in you.

❖

After 700 years Roger Bacon will have a statue erected to his memory at Oxford. Bringing home the bacon is a slow process sometimes.

❖

Petty pilferings are going on while soap is being shipped in New York for New Zealand, says a New Zealand paper. It alleges an alliance between dishonest cops and robber shipping gangs. Soap stealing sure does take the cake, but perhaps it is needed for another clean-up.

❖

Joseph J. O'Brien, secretary of the California State Association, is the founder and editor of an interesting little monthly called "Invention." It's meaty, wise and pertinent, and should interest even the layman.

❖

Miss Jane Addams, we have before remarked, ran second in a 10,000-name race with Thomas A. Edison as "the most useful citizen," and deservedly so. Now she is quoted as saying, "The fewer clothes we can wear, the better." Won't say another word until we hear from the lady.

❖

Seven hundred feet below the East River, two sections of the Catskill aqueduct tunnel met—and the variation was less than half an inch! Get that inch! No guesswork in this kind of engineering eh, what?

❖

New York's attorney-general says that electrical-plant employees are entitled to one day's rest in seven. Better talk with the boss, judge; some of us can't make him see it your way.

❖

Out at the pumping station the employees can't get a day a week, the law's agin' it, states this same judge. "Pumpers, pump, ding ye, and keep apumpin'!" as John Foote told the band at the Springfield convention.

❖

The Geological Survey again takes its pen in hand to remind us that gold is still worth $20.671834625323 per ounce. Thanks, fellers, thanks, but say, what does it look like, and feel like, to have and to hold an ounce of this metal?

❖

Down in Paraguay difficulty is found in successfully raising hogs. Come on up to New York, Jose and Miguel. You might work up a good shop kink by watching the mob on the elevated roads. These hogs run wild; don't need raising. And, then, you might give us a tip on how to improve the breed.

❖

At last we have official sanction to pin a rose on Father. The national House will make the first Sunday in June "Fathers' Day," Father to wear a rose. Mother used to "pin a rose on Me"; now, Father will make a stab at it.

❖

Are the Damms and the Hells going to start something? We have been somewhat facetious respecting the whole Damm family, but the Hells are new to our midst. Adam Damm (sort of tautological, what?) says the Hells are trying to outbid the Damms for public attention. Further, he says there are enough Damms to sink the Hells into oblivion; enough, in fact, to patch that celebrated mile.

❖

Waste Charged to Locomotive Heads—Headline.

Sounds like we must soon expect a head-on collision between some of the American Locomotive Co. officials. It is charged that the managers are running by signals and won't put on the brakes.

Meeting of Cotton Manufacturers

The National Association of Cotton Manufacturers held its 95th semiannual meeting at the Chalfonte, Atlantic City, N. J., on Sept. 30 to Oct. 2. It was attended by about 200 members. In addition to discussing the papers read, the meeting brought together many leading representatives of the cotton manufacturers for interchange of opinions on trade conditions and prospects.

The first session was called to order by President Edwin Farnham Greene, treasurer of the Pacific Mills and president of Lockwood, Greene & Co. Addresses of welcome were delivered by A. T. Bell, city clerk, in absence of Mayor Riddle, on behalf of the municipality, and by the Hon. Joseph W. Salus, president of the Business Men's League of Atlantic City, who said the association was "one of the most potent factors of National commerce and prosperity." Response was made by First Vice-President Albert Greene Duncan, treasurer of the Chicopee Manufacturing Co., and of Harmony Mills (Cohoes).

In his address President Greene stated that the textile industry is now entering a period of operation under conditions which exact most skillful mill management combined with the highest order of business administration. As to scientific management, the speaker said: "Any observant outsider in a casual visit to our mills may think he sees ways of economizing, but those who have been in the business realize that there are complex conditions to contend with.

Following the addresses, the president and the two guests held an informal reception which was also attended by the ladies.

On Wednesday and Thursday two daily sessions were held at which 14 papers were discussed, embracing a number of subjects which in ordinary mill organization are pertinent to power-plant management and operation.

In his paper on "Industrial Accidents, Their Compensation and Prevention," Carl M. Hansen, New York, said that "by establishing higher standards of safety and sanitation, you increase the efficiency of your workmen. All these exposed belts, shafting, gears, pulleys, clutches, couplings, setscrews, keys, pins, bad floors, and unprotected openings in floor, stairs without handrail, with bad treads and poorly lighted, insecure platforms and runways without rails and toe board, bad light throughout the plant; disorderly arranged machinery and débris littered throughout an establishment cause great loss in the productive efficiency of your employees."

L. H. Kunhardt, vice-president and engineer of the Boston Manufacturers' Mutual Fire Insurance Co., in his paper, "Preservation of Life in Factory Fires," stated that "the larger factories have given most serious consideration to the matter of self protection with the result that they have been almost entirely free from loss of life when fires occur. The most vital factor in preventing loss of life and property is the automatic sprinkler, and this alone will render the great majority of fires harmless to life and of small damage to property, but there must be secondary safeguards briefly covered by the general name 'slow-burning construction,' which means brick or equivalent walls, solid floors without openings between stories, stairs in outside towers, elevator and other openings protected and safe arrangement of processes. Then, even if very rarely sprinklers fail for any cause, the spread of fire will be so retarded that the operatives can escape without harm."

Speaking on the topic, "Methods of Cost and Clear Accounting Essential to Efficient Mill Management," Clinton H. Scovel said that the equipment in each production center represents an investment of capital; requires the payment of taxes and insurance; it suffers depreciation, and it incurs charges for power, repairs, such indirect items as superintendence, inspection and helpers' services. If the mill shuts down, the power may be shut off and the overseers and second hands dismissed; but so long as it runs, however shorthanded or inefficient, these charges do not change materially. Very few mills have calculated the burden element in cost with any degree of precision, and it is not necessary that a business should be approaching failure in order to have work of this kind highly profitable.

In a paper on "Developments in Lubrication," W. F. Parish, New York City, claimed that the rule is that within limits a reduction in the viscosity of a lubricant will bring about a reduction in the coefficient of friction, or a reduction in temperature and in lost power. Most of the resistance or lost power in running machinery is caused by the internal friction or the fluid friction in the lubricant used, assuming that an average condition with an average oil is under consideration. It is only occasionally, through the use of a very improper oil or lack of feed of a proper oil, that actual metal

wear on properly constructed bearings takes place. Wherever bearings are flooded, or where a proper working oil film is maintained, a very large proportion of the frictional loss is directly due to the internal fluid friction of the oil itself, and this can be influenced one way or another along certain prescribed lines.

In working entirely with paraffin base oils the coefficient of friction is influenced directly in proportion to the rise and fall of the viscosity of the oil used, always within the limits of speed and pressure. Working with oils made from the new Southern or Texas asphaltic base crudes the same general rule also applies.

"I have the honor," said Mr. Parish, "to present for the first time scientific and practical tests and investigations which show that, while the general law of the relation of viscosity to power is applicable to lubricants made from any one crude or classification of crude, a modification is necessary when tests on oils made from the paraffin base crudes are compared with tests on oils made from the new Texas base crudes. This technical feature of the Texas base oils is of the utmost importance, not only from a scientific, but from a practical standpoint.

"The great technical advantages in being able to use a heavier-bodied, higher-viscosity oil for lubrication are obvious. Less oil of this kind will be used, as it will not flow as rapidly from the oil can or cup. In the case of textile-mill work, a heavier-bodied oil will not drip as rapidly from the boxes or shafting. When the plant is closed down, the heavier-bodied oil will have a greater tendency to stay between the surfaces, so that in starting, the initial friction or power will be better influenced. The lighter-bodied oils at times when the machines are at rest, drip out or are pressed out between the surfaces, and high temperatures result upon starting the machines, owing to the fact that little or no oil is present between the surfaces to be lubricated during the start and until the oil boxes or cups are brought into operation. Where it is possible to use a heavier-bodied oil of higher viscosity, this effect is not as pronounced."

A paper on "Advantages of Commercial Motor Cars in Cotton Manufacture," by Day Baker, of Boston, Mass., was well illustrated by the stereopticon. The views showed the extensive use made of trackless electric-motor vehicle work for indoor and outdoor industrial transportation. Referring to the advantages of electric over other forms of motor vehicles, Mr. Baker said that while the current for the charging of the batteries is figured in all commercial estimates at 4c. per kw.-hr., most mills or factories can operate a small waterwheel or small engine during the night, furnishing power to charge the batteries, or taking current from the lighting plant before the peak of the load, utilizing capacity of a power equipment which would otherwise be idle and unproductive, thus reducing the cost of current to a small fraction of a cent per kilowatt-hour.

"Care of Belts," by Dudley Wilcox, Hartford, Conn., had an interesting reference to a scientific method of inspection of the tension of small belts. It also described a portable apparatus for determining the proper amount for cutting out of the length of belts for obtaining a tension which will be within the elastic limit of belt strength and not productive of unnecessary journal friction.

"It has been proved conclusively by a series of experiments lasting over nine years," continued Mr. Wilcox, "that tension of belts has more to do with their life than any other one factor.

"The average leather belt during its life stretches at least 6 per cent. of its original length, stretching rapidly from the beginning till a period of four to twelve months has elapsed, then a stretching almost uniformly for a number of years dependent upon the care it receives until finally its elasticity is practically gone, and the question arises whether it is cheaper to keep on repairing it every week or two or to buy a new belt. It has been found that belts well proportioned for the work required of them need tightening at intervals of two to six months.

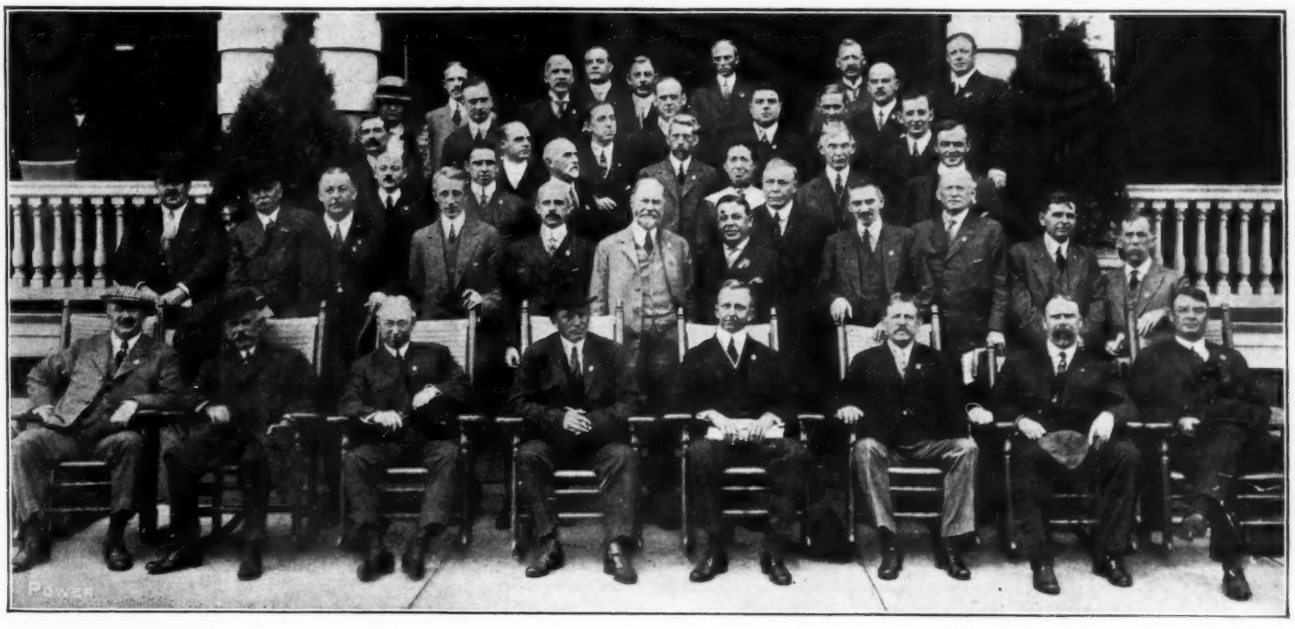
"Because of these conclusions and the usual lack of uniformity in the care of belts, a system and apparatus have been devised by means of which belts may be cared for in the most economical manner and interruptions to manufacture, due to failure of belting, may be reduced to a minimum.

"When this belt system was first applied in mills, a bench was built for weighing the tensions of belts by means of spring balances, and periodically all belts in the shop had to be brought to the bench for inspection and repair. With this inconvenience in mind, a portable device has been perfected for measuring the tensions of small belts while in place on their pulleys.

"It consists of three parallel steel rollers about 4 in. long by 1 in. in diameter, spaced about 10 in. apart and having one end of the axle bolt of each connected with a frame. The two end rollers have their axles fixed rigidly to the frame, while the spindle of the central roller is fastened to a slide on the frame and moves in a plane at right angles to the plane of the other two axles. This movement is controlled by a compression spring pressing the axle bolt of the movable roller against a cam revolved by a handle. When the handle of the apparatus is turned a weighing spring presses the middle roller gainst the belt, flexing it, and showing the tension of the belt on the graduated rod.

"By means of this device, the tensions of all belts up to 4 in. wide may be measured accurately while belts are in working position over their pulleys. With it belts may be inspected in about a quarter minute each and the amount necssary to be removed to bring the belt up to its proper maximum tension is indicated directly on the scale."

"Purchasing Coal Under Specifications" is the subject of a paper presented by George S. Pope, engineer-in-charge of government fuel inspection, Bureau of Mines, Washington, D. C., to be published in an early issue.



GROUP OF OFFICERS, MEMBERS AND GUESTS AT MEETING OF NATIONAL ASSOCIATION OF COTTON MANUFACTURERS, ATLANTIC CITY, N. J., OCT. 1, 1913

Illuminating Engineers Meet

The Illuminating Engineering Society held its seventh annual convention at Pittsburgh on Sept. 22 to 25, with headquarters at the Hotel Schenley. The meeting was called to order by Chairman C. S. Littlefield, and W. H. Stevenson, president of the Pittsburgh Chamber of Commerce, made an address of welcome to which President Millar and Vice-President Macbeth responded.

One of the features of the meeting was the presentation to the society by Prof. Hoadley, of a historic gavel symbolic in its design of the development of artificial illumination in this country up to the present time.

The report of the committee on progress completed the business of the morning session.

In the afternoon presentation of papers began and extended over Tuesday, Wednesday, Thursday and part of Friday. Following is a list of the papers presented:

"Recent Improvements in Incandescent Lamp Manufacture," by Ward Harrison and E. J. Edwards; "The Cooling Effect of Leading-In Wires Upon the Filaments of Lamps of the Street Series Type," T. H. Amrine; "Modern Practice in Street Illumination," S. G. Hibben; "Hospital Lighting," William S. Kilmer; "The Efficiency of the Eye Under Different Systems of Illumination—The Effect of Varying the Distribution and Intensity of Light," Dr. C. E. Ferree; "The Psychological Values of Light, Shade, Form and Color," Dr. F. Park Lewis; "Some Theoretical Considerations of Light Production," W. A. Darrah; "The Use of Nitrogen at Low Pressure in Tungsten Lamps," G. M. J. Mackay; "The Neon Tube Lamp," M. George Claude; "The Quartz Mercury Vapor Lamp and Its Application," W. A. D. Evans; "The Evolution of the Lamp," Roscoe E. Scott; "Description and Demonstration of the Lighting of Soldiers' Memorial Hall"; "Church Lighting," R. B. Ely; "Experiments in the Illumination of a Sunday School Room with Gas," Edwin F. Kingsbury; "Distinctive Store Lighting," C. L. Law and A. L. Powell; "Factory Lighting," M. H. Flexner and A. O. Dicker; "Some Commercial Aspects of Gas Lighting," J. E. Philbrick; "The Lighting of Show Windows," H. B. Wheeler; "The Illuminating Engineering Laboratory of the General Electric Company at Schenectady," S. L. E. Rose; "The Pentane Lamp as a Working Standard," E. C. Crittenden and A. H. Taylor; "The Photo-Electric Cell in Photometry," Prof. F. K. Richtmyer; "Some Studies in Accuracy of Photometry," Evan J. Edwards and Ward Harrison; "Characteristics of Enclosing Glassware," Van Rensselaer Lansingh.

On Wednesday evening a banquet was held at the Hotel Schenley. G. H. Harries acted as toastmaster, and among the speakers were President P. S. Millar, President-elect C. O. Bond, C. M. Bregg and J. A. Brashear.

The present membership of the society is 1350.

SOCIETY NOTES

Pratt Institute, Brooklyn, N. Y., has opened an evening course in trade teaching. The directors believe that there is a strong demand for men with trade experience as teachers in industrial and trade schools. Particulars may be had by addressing Director Samuel S. Edmands.

At a regular meeting of Minneapolis Association No. 2, Minneapolis, of the National Association of Stationary Engineers on Sept. 28, State President-elect H. F. Mueller was installed by State Deputy T. S. F. Hayes. The recently returned delegates to the national convention at Springfield, Mass., gave interesting reports of that event, and President Morris, on behalf of the delegates, formally presented the association with the silk altar flag awarded it for the large increase in its membership during the past year.

OBITUARY

JOHN F. ENSIGN

On Sept. 25 John F. Ensign, chief of the division of locomotive boiler inspection of the Interstate Commerce Commission, died in Washington, D. C., at the age of 51 years. His wife and daughter accompanied the body to Denver, Colo.

Mr. Ensign was the first man to fill the position of boiler inspector, having been named for that place July 1, 1911, by President Taft, the day the law went into effect. Mr. Ensign was born in Marathon, N. Y.

PERSONALS

O. W. Watson has been appointed superintendent of the Leicester plant of the Worcester (Mass.) Electric Light Co.

E. M. Westhoven has been appointed superintendent of the Woodlake station of the Mount Whitney Power & Electric Co., of Visalia, Calif.

Frank J. Corbett has opened an office at 30 Church St., New York City, for the sale of condensers, pumps, cooling towers and other apparatus used in the power field.

S. M. Udale, formerly assistant research engineer with the Studebaker Corporation, is now in charge of the laboratory and motor-testing plant of Joseph Tracy, whose New York office is in the United States Rubber Building, 1790 Broadway, New York City.

Joseph Harrington has resigned his position as chief consulting engineer for the Green Engineering Co. and engaged in independent practice. He will specialize in boiler-plant work, such as improving plant design, equipment and operation and increasing organization efficiency. Mr. Harrington was connected with the Green Engineering Co. for 12 years.

Thomas R. Armstrong, of Boston, recently appointed a number of the Massachusetts Board of Boiler Rules to represent the operating engineers of the state, was selected and recommended to the governor by the New England Society of Superintending Engineers, whose members are the chief engineers of the large plants of the state. The society also recommended the appointment of George Luck, the new chief inspector of the boiler-inspection department. Mr. Armstrong is an engineer of large practical experience, having been a commissioned engineer in the United States revenue service, an engineer officer in the United States Navy, and a chief engineer of ocean-going vessels in the merchant service. He has also held the positions of chief engineer of Boston public institutions; of the Sears estate, Boston; of the Mason Machine Works, Taunton, Mass.; an inspector of engineering materials for the Japanese government at the Cramps shipyard, at Philadelphia; superintendent of construction for the C. E. Cotting estate, Boston. Mr. Armstrong is at present chief engineer of the Boston Safe Deposit & Trust Co.; chief engineer-in-charge of the U. S. S. "Chicago," "Rodgers" and the "Macdonough," which vessels are attached to the Massachusetts Naval Brigade. He holds the rank of lieutenant-commander. Much credit is due the officers and members of the New England Society of Superintending Engineers in selecting an engineer so well qualified to represent the engineers of Massachusetts on the Board of Boiler Rules.

The National Tube Co.'s bulletin No. 16 deals with tests applied to boiler tubes. The manner in which the tubes are tortured is shown by numerous halftone engravings, and a table of dimensions, weight, and test pressures is given. The bulletin is worth sending for. It may be had free upon application to the Pittsburgh office of the company.

STATEMENT OF THE OWNERSHIP, MANAGEMENT, CIRCULATION, ETC.

of Power, published weekly at New York, N. Y., required by the Act of August 24, 1912.

Editor, Fred R. Low, New York, N. Y.
Managing Editor, H. R. Cobleigh, New York, N. Y.
Business Manager, William Buxman, New York, N. Y.
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Owners of 1% or more of Stock Issued.

John A. Hill, 505 Pearl St., New York, N. Y.
Fred R. Low, 505 Pearl St., New York, N. Y.
John McGhie, 505 Pearl St., New York, N. Y.
Fred S. Weatherby, 1600 Beacon St., Boston, Mass.
Frederick A. Halsey, 505 Pearl St., New York, N. Y.
G. Eugene Sly, 50 Union Sq., New York, N. Y.
Frederick W. Gross, 447 E. 11th St., Erie, Pa.
Alfred E. Kornfeld, 114 E. 71st St., New York, N. Y.
Emma B. Hill, 80 Munn Ave., E. Orange, N. J.

The balance of the stock issued (less than 1% each) is owned by 62 employees, one ex-employee, and 11 others who are wives, daughters or relatives of employees.

There are no bondholders, mortgagees, or other security holders.

C. W. Dibble, Vice-President,
HILL PUBLISHING COMPANY.

Sworn to and subscribed before me this 29th day of September, 1913.

RICHARD L. MURPHY,
Notary Public.
(My commission expires March 30, 1915)